*PNS SCHOOL OF ENGINEERING &TECHNOLOGY*

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**DEPARTMENT OF MECHANICAL ENGINEERING**

# LECTURER NOTES

**Year & Semester:** **3rd Year, 5th- Semester**

**Subject Code/Name: TH-5,REFRIGERATION & AIR CONDITIONING**

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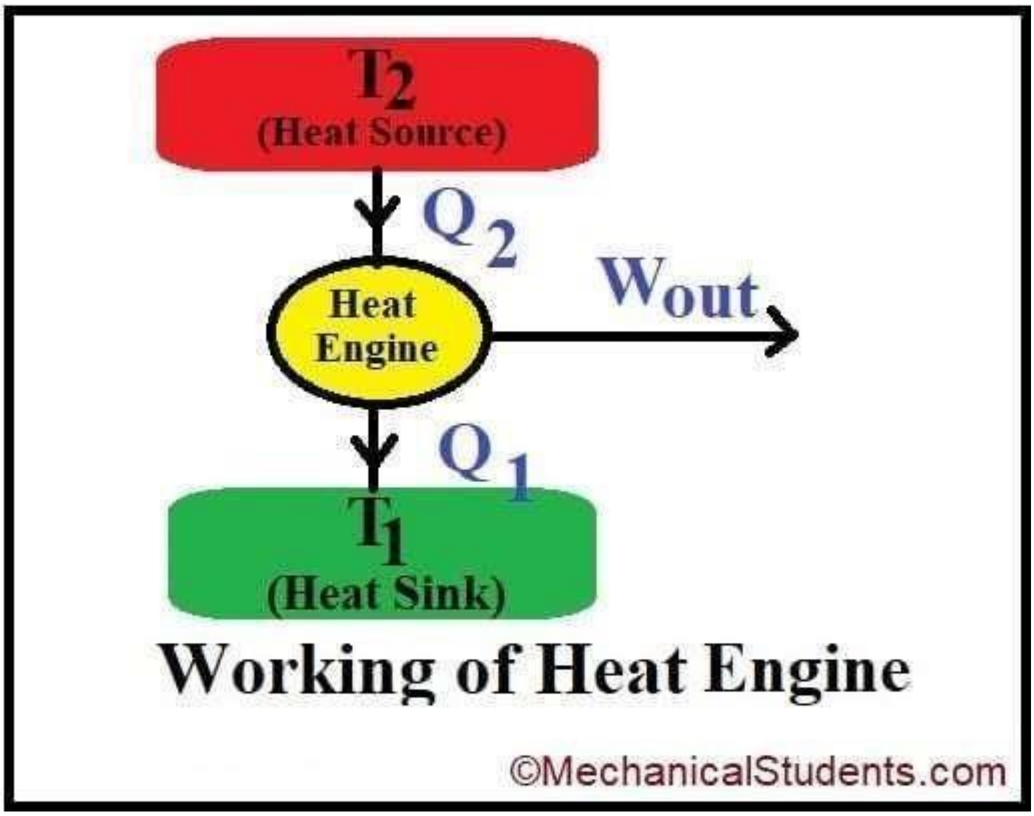
**Refrigeration**

Refrigeration is defined as the process of achieving and maintaining a temperature below that of the surroundings. The aim is to cool some product or space to the required temperature.

Air Conditioning refers to the treatment of air and to simultaneously control its temperature, moisture content, cleanliness, odour and circulation, as required by occupants, a process, or products in the space.

## Difference between a Refrigerator, Heat Pump, and Heat Engine

A heat engine is a system that converts Thermal energy into Mechanical Energy.



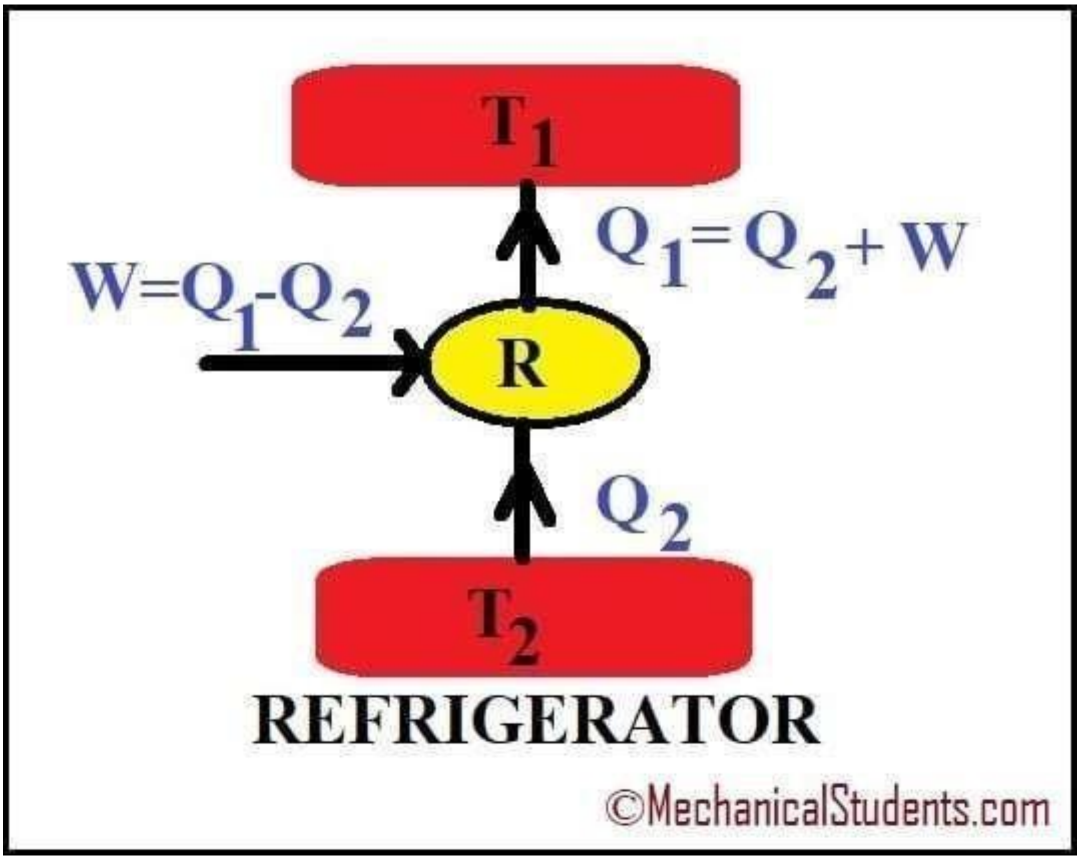

In a heat engine, the heat supplied to the engine is converted into useful work. If **Q2** is the heat supplied to the engine and **Q1** is the heat rejected from the heat engine, then the **network done** by the engine is given by

**We = Q2-Q1**

So the performance of the engine or Efficiency is given by

Generally, **Efficiency** is calculated as = **We/Q**

A refrigerator is a reversed heat engine, where heat is pumped from low temperature (cold body--> Q1) to high temperature (hot body-->Q2)

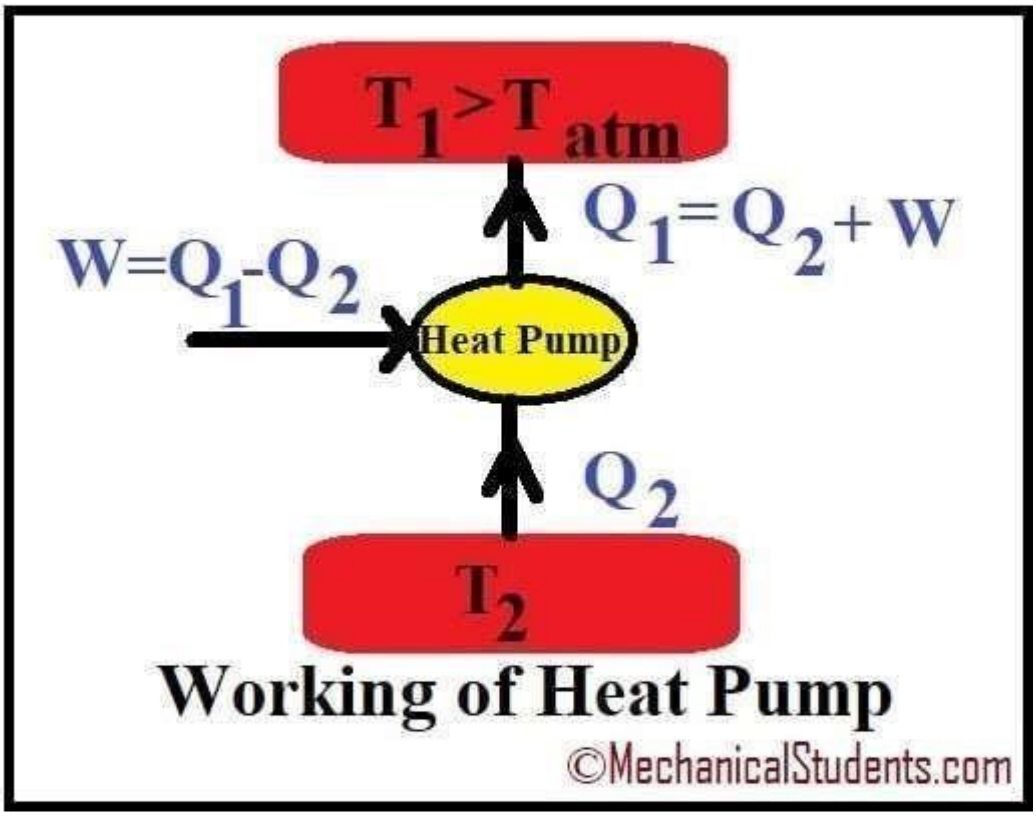



So, Work **WR** is required to be done on the system.

**WR = Q2-Q1**

The performance of a refrigerator is the "ratio of the amount of heat taken from the Cold body Q1 to the amount of work to be done on the system WR.

Any refrigerating system is a heat pump, which extracts heat from a cold body and delivers it to a hot body.




Thus there is no difference in the operation cycle of a refrigerator and a heat pump.

- The main difference between them is in their operating temperatures.
- A refrigerator works between cold body temperature (T1) and atmospheric temp (Ta) whereas the heat pump operates between hot body temp (T2) and the atmospheric temperature (Ta).
- A refrigerator used for cooling in summer can be used as a heat pump for heating in the winter season.
- so **Wp = Q2-Q1**

Performance of refrigerator and heat pump

COP(Coefficient of performance)

COP is defined as the relationship between the power (kW) that is drawn out of the heat pump as cooling or heat, and the power (kW) that is supplied to the compressor.

the C.O.P is the reciprocal of efficiency and is given as

**(C.O.P)R = Q1/ WR = Q1 / (Q2-Q1) ------------ for refrigerator**

**(C.O.P)hp = Q2/ WR = Q2 / (Q2-Q1)----------- for heat pump**

| Refrigerator | Heat Pump | Heat Engine |
|---|---|---|
| A refrigerator is a reversed heat engine, where heat is pumped from a body at low temperature to a body at high temperature. | Any refrigerating system is a heat pump, which extracts heat from a cold body and delivers it to a hot body. | A heat engine is a system which converts Thermal energy into Mechanical Energy. |
| The **network done** by the refrigerator is given by W**R= Q2-Q1** | The **network done** by the heat pump is given by W**p= Q2-Q1** | The **network done** by the engine is given W**e= Q2-Q1** |
| The C.O.P. of Refrigerator is **(C.O.P)R = Q1/ WR = Q1 / (Q2-Q1)** | The C.O.P. of heat pump is **(C.O.P)hp = Q2/ WR = Q2 / (Q2-Q1)** | The C.O.P. of heat engine is **(C.O.P)e= (Q2-Q1)/Q2** |

Unit of refrigeration

Rating for Refrigeration indicates the rate of removal heat. The unit of refrigeration is expressed in terms of ton of refrigeration (TR). One ton of refrigeration is defined as the amount of refrigeration effect (heat transfer rate) produced during uniform melting of one ton (1000kg) of ice at 0°C to the water at the 0°C in 24 hours.

Calculation for one ton of refrigeration

Latent heat of ice is 335KJ/kg (heat absorbed during melting of one kg ice)
1 Ton of refrigeration, 1TR= 1000*335 in 24 hours

=(1000×335)/(24×60) in one minute
= 232.6 kJ/min

Theoretically one Ton of refrigeration taken as 232.6kJ/min, in actual practice, it is taken as 210kJ/min.
1 ton of refrigeration approximately equal to 3.5kw

# CHAPTER-2

Air cycle refrigeration is one of the earliest methods used for cooling. The key features of this method is that, the refrigerant air remain gaseous state throughout the refrigeration cycle. Based on the operation, the air refrigeration system can be classified into

1. Open air refrigeration cycle
2. Closed refrigeration cycle

**Open air refrigeration cycle**

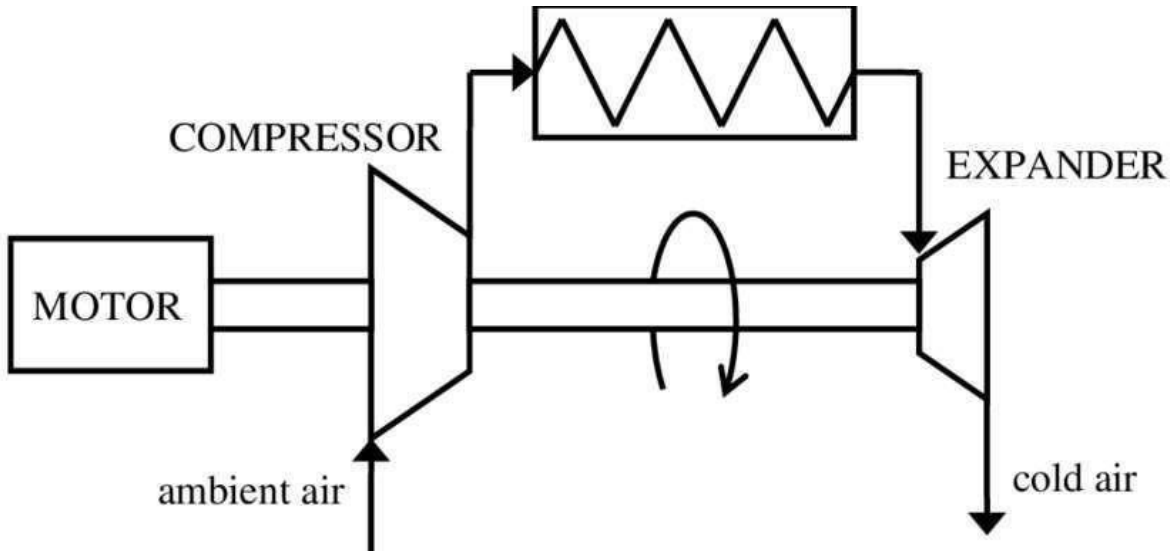
In an open refrigeration system, the air is directly passed over the space is to be cooled, and allowed to circulate through the cooler. The pressure of open refrigeration cycle is limited to the atmospheric pressure. A simple diagram of the open-air Refrigeration system is given below.

**Advantages and application**

- It eliminates the need of a heat exchanger.
- It is used in aircraft because it helps to achieve cabin pressurization and air conditioning at once

***Disadvantages***

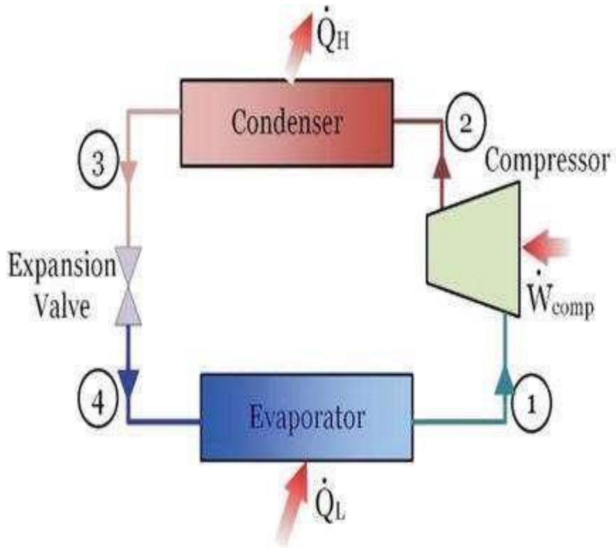
One of the disadvantages of this system is that its large size. The air supplied to the refrigeration system is at atmospheric pressure, so the volume of air handled by the system is large. Thus the size of compressor and expander also should be large. Another disadvantage of the open cycle system is that the moisture is regularly carried away by the circulating air, this leads to the formation of frost at the end of the expansion process and clogs the line, and hence a use of dryer is preferable to the open air refrigeration system.

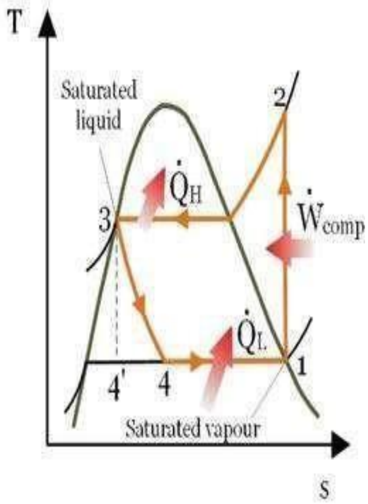



Open air refrigeration system

## Closed refrigeration system / Dense air refrigeration cycle

In closed or dense air refrigeration cycle, air refrigerant is contained within pipes and component part of the system at all time. The circulated air does not have to direct contact with the space to be cooled. The air is used to cool another fluid (brine), and this fluid is circulated into the space to be cooled. So the disadvantages listed in open air refrigeration can be eliminated. The advantages of closed air refrigeration system are listed below.




***Advantages***

- The suction to the compressor may be at high pressure, therefore the volume of air handled by the compressor and expander is low when compared to an open system. Hence the size of compressor and expander is small compared to the open air system.
- The chance of freezing of moister and choke the valve is eliminated.
- In this system, higher coefficient of performance can be achieved by reducing operating pressure ratio.

# Air Refrigerator Working On Bell-Coleman Cycle with PV and TS Diagram (Reversed Brayton or Joule Cycle)

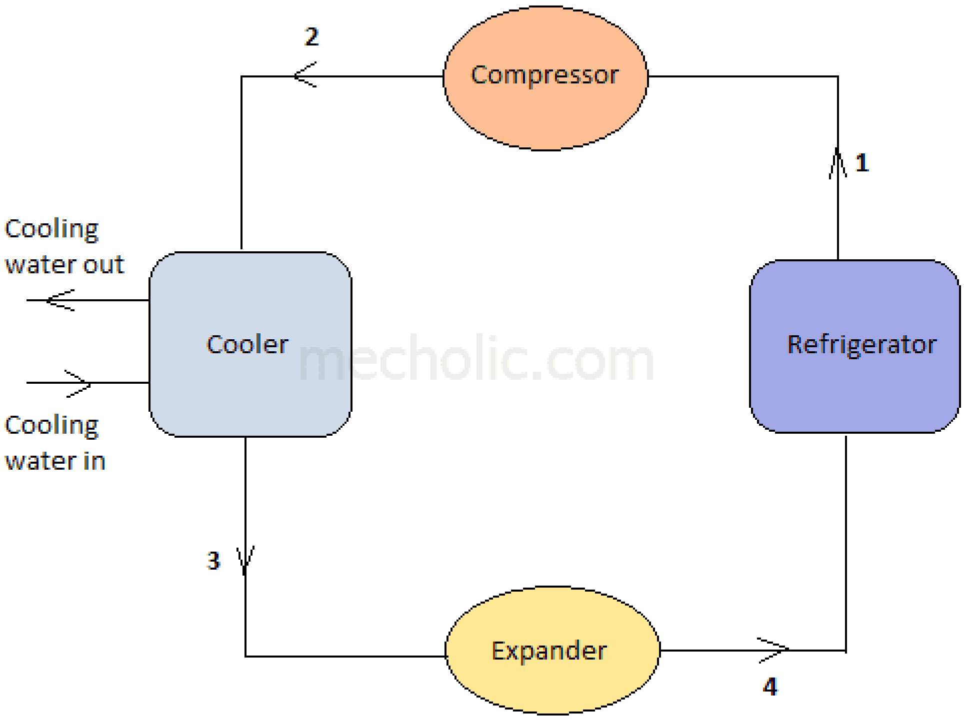
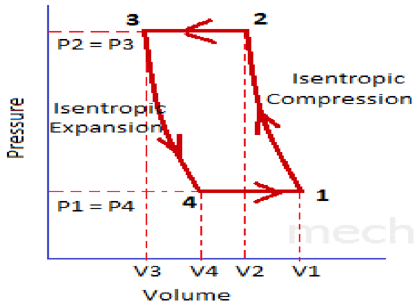
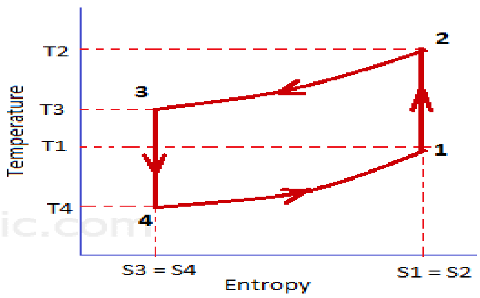



Fig shows a schematic diagram of Bell-Coleman refrigerator (reverse Brayton or joule cycle). This refrigeration system components consists of a compressor, cooler, Expander, and refrigerator. In this process, heat absorption and rejection follows at the constant pressure; the compression and expansion of process are isentropic.

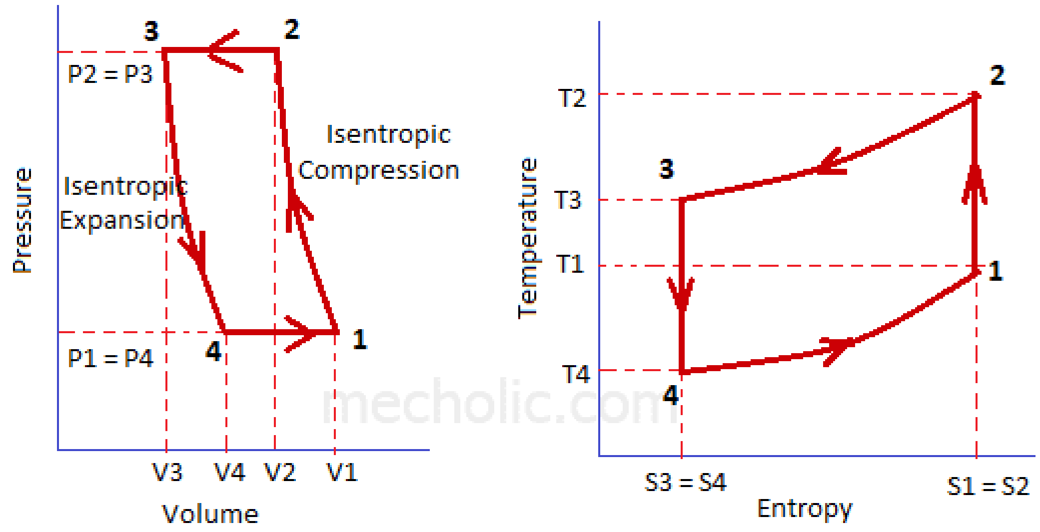
## Process in Bell-Coleman refrigeration




(a) P-V Diagram




(b) T-S Diagram




(a) P-V Diagram (b) T-S Diagram

Fig show P-V and T-S diagram of bell coleman refrigerator. Here $P_1$, $V_1$, $T_1$, $S_1$ represents the pressure, volume, temperature, entropy of air respectively at point 1. And so on. It represents the corresponding condition of air when it passed through the component.

1-2: Isentropic Compression
The Air drawn from refrigerator to air compressor cylinder where it compressed isentropically (constant entropy). No heat transfer by the air. During compression, the volume decreases while the pressure and temperature of air increases.

2-3: Constant pressure cooling process.
The warm compressed air is then passed through cooler, where it cooled down at constant pressure. The heat rejected per kg of air during this process is equal to
$q_{2-3} = Cp(T_2-T_3)$

3-4: isentropic expansion
No heat transfer takesplcae. The air expands isentropically in expander cylinder. During expansion, the volume increases, Pressure P3 reduces to $P_4$. ($P_4$ = atmospheric pressure). Temperature also falls during expansion from $T_3$-$T_4$.

4-1: Constant pressure expansion
Heat transfer from the refrigerator to air. The temperature increases from $T_4$ to $T_1$. Volume increases to $V_4$ due to heat transfer. Heat absorbed by air per kg during this process is equal to
$q_{4-1} = Cp(T_1-T_4)$

## Equation of Coefficient of performance (COP) of Bell Coleman cycle

Heat absorbed during cycle per kg of air $q_{4-1} = Cp(T_1-T_4)$
Heat rejected during cycle per kg of air $q_{2-3} = Cp(T_2-T_3)$
Then the work done per kg of air during the cycle is = Heat rejected – Heat absorbed

$= Cp(T_2-T_3) - Cp(T_1-T_4)$

Coefficient of performance,

$$C.O.P. = \frac{Heat\ absorbed}{Work\ done} = \frac{C_p(T_1 - T_4)}{C_p(T_2 - T_3) - C_p(T_1 - T_4)}$$

$$= \frac{(T_1 - T_4)}{(T_2 - T_3) - (T_1 - T_4)}$$

$$C.O.P. = \frac{T_4(\frac{T_1}{T_4} - 1)}{T_3(\frac{T_2}{T_3} - 1) - T_4(\frac{T_1}{T_4} - 1)} \qquad \text{(i)}$$

For isentropic compression process 1-2

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \qquad \text{(ii)}$$

For isentropic expansion process 3-4

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} \qquad \text{(iii)}$$

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Since, P2 = P3 and P1 = P4, therefore from equation (ii) and (iii)

Substitute equation (iv) in (i)

$$C.O.P. = \frac{T_4}{T_3 - T_4} = \frac{1}{\frac{T_3}{T_4} - 1}$$

$$= \frac{1}{\left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} - 1} = \frac{1}{\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1}$$

$$C.O.P. = \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}} - 1}$$

$$r_p = Compression\ or Expansion\ ratio = \frac{P_2}{P_1} = \frac{P_3}{P_4}$$

# Questions & Answers Ch-1 & 2

## Define refrigeration ?

The term **refrigeration** means cooling a space, substance or system to lower and/or maintain its temperature below the ambient one (while the removed heat is rejected at a higher temperature).[1][2] In other words, refrigeration is artificial (human-made) cooling.[3][4] Energy in the form of heat is removed from a low-temperature reservoir and transferred to a high-temperature reservoir.

## Define unit of refrigeration ?

The **unit of refrigeration** is expressed in terms of ton of **refrigeration** (TR). ... One ton of **refrigeration** is defined as the amount of **refrigeration** effect (heat transfer rate) produced during uniform melting of one ton (100kg) of ice at 0°C to the water at the 0°C in 24 hours.

## Define cop ?

The **coefficient of performance** or COP (sometimes CP or CoP) of a heat pump, refrigerator or air conditioning system is a ratio of useful heating or cooling provided to work required. Higher COPs equate to lower operating costs.

## Define refrigerating effect ?

Refrigeration effect is the amount of heat that each pound of refrigerant retains from the refrigerated space to deliver helpful cooling. In the gas cycle, the refrigeration effect is equivalent to the result of the particular warmth of the gas and the ascent in temperature of the gas in the low temperature side.

## Define open and closed air refrigeration system ?

**Open air refrigeration cycle:** When cooled air from the turbine enters the cabin and comes in physical contact with the occupants. It is not much in use because of moisture added to air in the cabin.

**Closed air refrigeration cycle OR dense cycle:** When cooled air from the turbine passes through the coil and a fan circulates and recirculates cabin air over it. The pressure of cooled air in such systems is much higher than in the open system. Because of high pressure, volume is less and hence density of air is high. It is therefore also called a dense system. It reduces compression ratio and hence COP is high. There is no moisture problem too.

**LONG QUESTIONS**

Q-Derive COP of Bell Coleman cycle.

Q_Describe different of different components of simple vapour compression cycle.

Q-describe different processes of SVCC with T-S & P-H Graph,

Q-Derive COP of SVCC with dry vapour after compression,

# CHAPTER-3 VAPOUR ABSORPTION SYSTEM

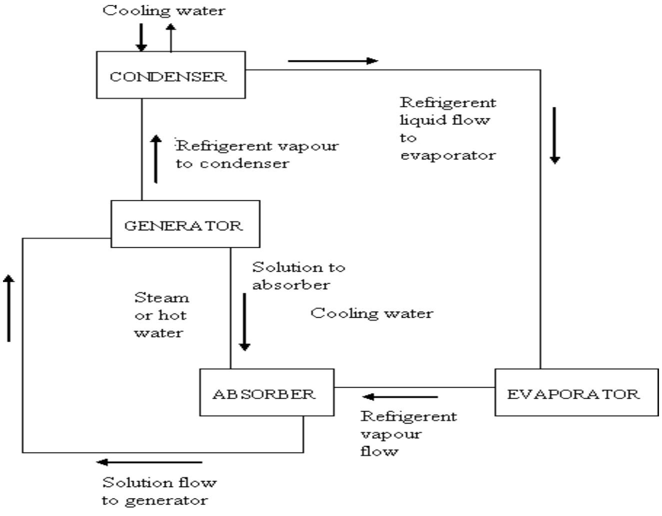
The vapour absorption refrigeration is heat operated system. It is quite similar to the vapour compression system.

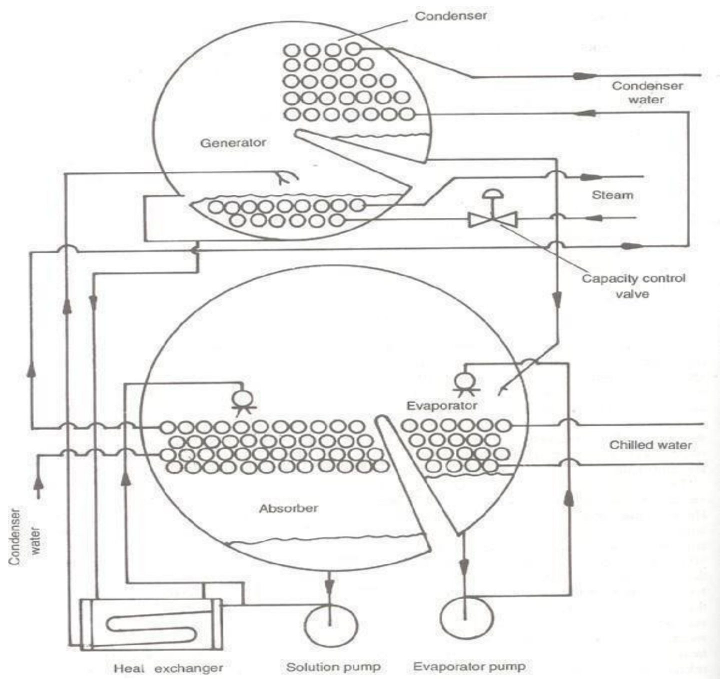
## VAPOUR ABSORPTION SYSTEMS

In both the systems, there are evaporator and condenser. The process of evaporation and condensation of the refrigerant takes place at two different pressure levels to achieve refrigeration inboth the cases. The method employed to create the two pressure levels in the system for evaporationand condensation of the refrigeration makes the two processes different. Circulation of refrigerant inboth the cases is also different.

In the absorption system the compressor of the vapour compression system is replaced by the combination of „absorber‟ and „generator‟. A solution known as the absorbent, which has anaffinity for the refrigerant used, is circulated between the absorber and the generator by a pump (solution pump). The absorbent in the absorber draws (or sucks) the refrigerant vapour formed in the evaporator thus maintaining a low pressure in the evaporator to enable the refrigerant to evaporate at low temperature. In the generator the absorbent is heated. There by releasing the refrigerant vapour (absorbed in the absorber) as high pressure vapour, to be condensed in the condenser. Thus the suction function is performed by absorbent in the absorber and the generator performs the function of the compression and discharge. The absorbent solution carries the refrigerant vapour from the low side (evaporator– absorber) to the high side (generator-condenser). The liquefied refrigerant flows from the condenser tothe evaporator due to the pressure difference between the two vessels; thus establishing circulation of the refrigerant through the system.

The absorbent solution passing from the generator to the absorber is hot and ha to be cooled. On theother hand the absorbent solution sent to the generator is cooled and has to be heated in the generator for the regeneration of the refrigerant. A shell and tube heat exchanger is introduced between the generator and the absorber.




Schematic Diagram of Absorption System of

Refrigeration Schematic Sketch of a Lithium-Bromide Absorption

Machine – Single Stage

There is number of vapour absorption system depending on the absorbent e.g. ammonia absorbent system, lithium bromide absorption system etc. Ammonia absorbent systems were used in the early stages of refrigeration. This system uses ammonia as the refrigerant and water.

## ***Short and long questions ch-3***

1-a-what is the function of analyser?

b-what is the function of generator in VARS?

c-what is weak solution and strong solution in VARS?

d-what is the function of heat exchanger in between evaporator & absorber.

e-what is strong solution in VARS?

f-What is the function of rectifier ?

2-a-Write the advantages of VARS over compression refrigeration system.

b-With a neat diagram discuss the working of practical VARS.

# CHAPTER 4 REFRIGERATION EQUIPEMENTS

## COMPRESSORS

### 3.2.1 Types of Compressor

There are different types of compressors that generally used in industry are,

(a) Reciprocating compressor

(b) Centrifugal compressor

(c) Rotary compressor

(d) Screw compressor

(e) Scroll compressor

The reciprocating and screw compressors are best suited for use with refrigerants which require a relatively small displacement and condense at relatively high pressure, such as R-12, R-22, Ammonia, etc.

The centrifugal compressors are suitable for handling refrigerants that require large displacement and operate at low condensing pressure, such as R-11, R-113, etc.

The rotary compressor is most suited for pumping refrigerants having moderate or low condensing pressures, such as R-21 and R-114; this is mainly used in domestic refrigerators.

**Reciprocating Compressor**

The compressors in which the vapour refrigerant is compressed by the reciprocating (i.e. back and forth) motion of the piston, called reciprocating compressors. These compressors are used for refrigerants which have comparatively low volume per kg and a large differential pressure, such as ammonia, R-12, R-22, etc.

*Basic Cycle for Reciprocating Compressor*

The p-v diagram of a reciprocating compressor is shown in the Figure 3.1 along with the skeleton diagram of the cylinder and piston mechanism.

When the piston is in the extreme left position of the inner dead centre (IDC), the volume occupied by the gas is $V_c = V_3$ called clearance volume, i.e. the volume between the piston and cylinder head. As the piston moves outward, the clearance gas expands to 4, when the pressure inside the cylinder is equal to the pressure at the suction flange of the compressor. As the piston moves further, the suction valve S opens and the vapour from the evaporator is sucked in till the extreme right position of the outer dead centre (ODC) is reached. At this position the volume occupied by the gas is $V_1$. The stroke or swept volume or piston displacement is

$$V_p = (V_1 \quad V_3) \quad \pi D^2$$

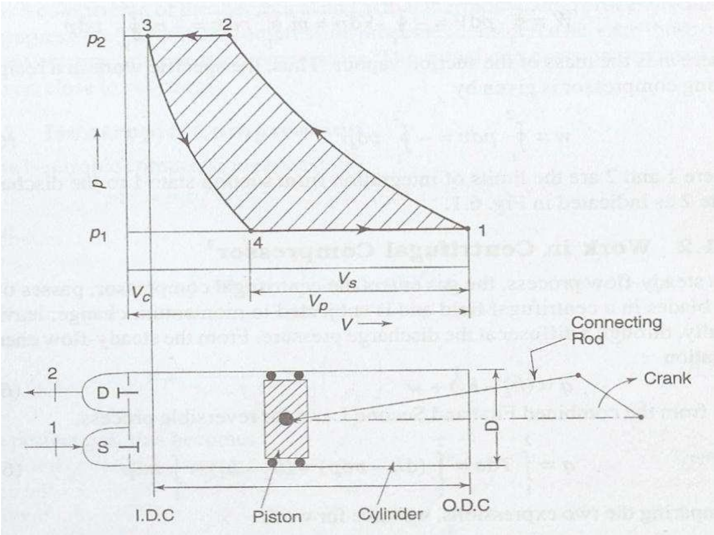



**Figure 3.1 : Cylinder and Piston Mechanism and P-V Diagram of a Reciprocating Compressor**

Where *D* is the bore or diameter and *L* is the stroke, i.e. the distance traveled by the piston between IDC and ODC of the cylinder. At 1, the suction valve closes as the piston moves inwards and the compression begins. At 2, the pressure in the cylinder is equal to the pressure at the discharge flange of the compressor. A further movement of the piston inward results in the pressure in the cylinder exceeding the condenser pressure. This opens the discharge valve *D* and the vapour from the cylinder flows into the condenser till the piston reaches again the IDC position. Gas equal to the clearance volume *Vc* remains in the cylinder and the cycle is operated.

The work done for compression is given by the cyclic integral of $pdV$.

Hence, $$W = \oint pdV = \int_1^2 pdV + \int_2^3 pdV + \int_3^4 pdV + \int_4^1 pdV$$

$$= \int_1^2 pdV + p_2(V_3 - V_2) + \int_3^4 pdV + p_1(V_1 - V_4)$$

= Area 1-2-3-4

It will be seen that this area is also expressed by the term $-\oint Vdp$. Hence

$$W = \oint pdV = -\oint Vdp = -m\oint vdp$$

where, *m* is the mass of the suction vapour. Thus, the specific work in a reciprocating compressor is given by

$$w = -\int vdp$$

*Volumetric Efficiency of Reciprocating Compressor*

Volumetric efficiency is the term defined in the case of positive displacement compressors to account for the difference in the displacement in-built in the compressor *Vp* and actual volume *Vs*, of the suction vapour sucked and pumped. It is expressed by the ratio

$$\eta_v = \frac{V_s}{V_p} \qquad \ldots 3.2$$

*Clearance Volumetric Efficiency*

The clearance or gap between the I.D.C. position of the piston and cylinder head is necessary in reciprocating compressors to provide for thermal expansion and machining tolerances. A clearance of (0.005L+0.5) mm is normally provided. This space together with the volume of the dead space between the cylinder head and valves, forms the clearance volume. The ratio of the clearance volume $V_c$ to the swept volume $V_b$ is called the clearance factor C, i.e.,

$$C = \frac{V_c}{V_p} \qquad \ldots 3.3$$

This factor is normally ≤ 5 per cent.

The effect of clearance in reciprocating compressors is to reduce the volume of the sucked vapour, as can be seen from Figure 3.1. The gas trapped in the clearance space expands from the discharge pressure to the suction pressure and thus fills a part of the cylinder space before suction begins. Considering only the effect of clearance on volumetric efficiency, we have from Figure 3.1 , for clearance volumetric efficiency

$$\eta_{cv} = \frac{V_1 - V_4}{V_p} = \frac{\left(V_p \ V_c\right) - V_4}{V_p} \qquad \ldots 3.4$$

The volume occupied by the expanded clearance gases before suction begins is

$$V_4 = \left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}} V_c = \left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}} VC_p \qquad \ldots 3.5$$

so that

$$\eta_{cv} = \frac{V_p + CV_p - CV_p\left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}}}{V_p}$$

$$= 1 + C - C\left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}} \qquad \ldots 3.6$$

*Variation of Volumetric Efficiency with Suction Pressure*

As shown in Figure 3.2 the nature of variation of the p-V diagram of a reciprocating compressor with suction pressure for constant discharge pressure. It is seen that with decreasing suction pressure, or increasing pressure ratio, the suction volume V and hence volumetric efficiency decrease until both become zero at a certain low pressure p'. Thus the refrigerating capacity of a reciprocating compressor tends to zero with decreasing evaporator pressure.

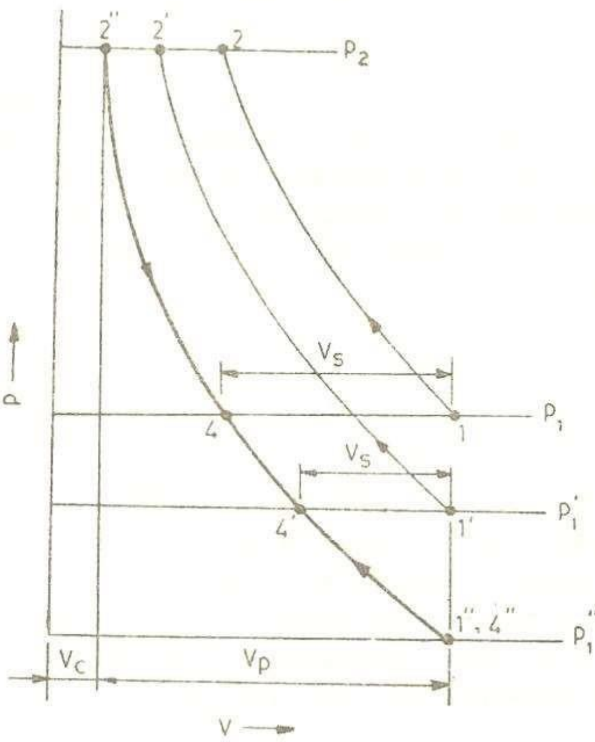



**Figure 3.2: Decrease in Suction Volume in a Reciprocating Compressor with Decreasing Evaporator Pressure**

*Effect of Valve Pressure Drops*

For the flow of any fluid, the pressure must drop in the direction of flow. Both suction and discharge valves will open only when there is a pressure drop across them. The effect of these pressure drops on the indicator diagram of the compressor is shown in Figure 3.3 It is seen that as a result of throttling or pressure drop on the suction side the pressure inside the cylinder at the end of the suction stroke is Ps while the pressure at the suction flange is $P_1$. The pressure in the cylinder rises to the suction flange pressure $P_t$ only after the piston has travelled a certain distance inward during which the volume of the fluid has decreased from $\left(V_p + V_c\right)$ to $V_1$.

Assuming the compression index to be n instead of $\gamma$, as the compression process is also polytropic due to heat exchange with cylinder walls and friction, we have

$$V_1 = \left(V_p + V_c\right)\left(\frac{p_s}{p_1}\right)^{\frac{1}{n}} \qquad \ldots 3.8$$

The expression for volumetric efficiency becomes

$$\eta_{cv} = \frac{V_1 - V_4}{V_p} = \frac{\left(V_p - V_c\right)\left(\frac{p_s}{p_1}\right)^{\frac{1}{n}} - V_c\left(\frac{p_2}{p_1}\right)^{\frac{1}{m}}}{V_p}$$

$$= \left(1 + C\right)\left(\frac{p_s}{p_1}\right)^{\frac{1}{n}} - C\left(\frac{p_2}{p_1}\right)^{\frac{1}{m}} \qquad \ldots 3.9$$

Considering the effect of pressure drop at the discharge valve as well, it can be shown that the expression for volumetric efficiency is

$$\eta_{cv} = (1+C)\left(\frac{p_s}{p_1}\right)^{\frac{1}{n}} - C\left(\frac{p_d}{p_1}\right)^{\frac{1}{m}} \qquad \text{...3.10}$$

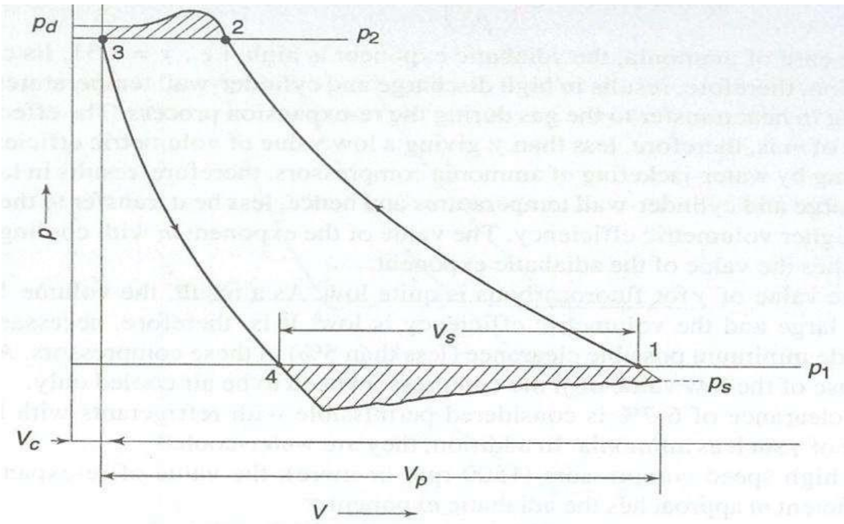



**Figure 3.3: Effect of Valve Pressure Drops**

*Overall Volumetric Efficiency*

Considering the effect of wire-drawing at the valves, polytropic compression, re expansion, and leakage, we may write the expression for the overall or total volumetric efficiency as follows

$$\eta_v = (1+C)\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} - C\left(\frac{p_d}{p_1}\right)^{\frac{1}{m}} - 0.01r \qquad \text{...3.11}$$

The methods of improving the volumetric efficiency include the following:

(a) Providing clearance as small as possible,

(b) Maintaining low pressure ratio,

(c) Cooling during compression,

(d) Reducing pressure drops at the valves by designing a light-weight valve mechanism, minimizing valve overlaps and choosing suitable lubricating oils.

*Effect of Clearance on Work*

The effect of the clearance volume on the work of compression is mainly due to the different values of the exponents of the compression and expansion processes. If the exponents are different, the net work is given by

$$W = -\int_1^2 V dp + \int_4^3 V dp$$

$$= -\frac{n}{n-1} p_1 V_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] + \frac{m}{m-1} p_1 V_4 \left[\left(\frac{p_2}{p_1}\right)^{\frac{m-1}{m}} - 1\right] \qquad \text{...3.12}$$

When the two exponents are equal, i.e. m= n

$$W = \frac{n}{n-1} p_1 V_s \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] \qquad \text{...3.13}$$

where $V_s = V_1 - V_4$ volume of the vapour sucked. Thus the work is only proportional to the suction volume. The clearance gas merely acts like a spring, alternately expanding and contracting. In practice, however, a large clearance volume results in a low volumetric efficiency and hence large cylinder dimensions, increased contact area between the piston and cylinder and so, increased friction and work.

## Centrifugal Compressor

A single-stage centrifugal compressor mainly consists of the following four components as shown in Figure 3.4.

(a) An inlet casing to accelerate the fluid to the impeller inlet.

(b) An impeller to transfer energy to the fluid in the form of increase in static pressure and kinetic energy.

(c) A diffuser to convert the kinetic energy at the impeller outlet into pressure energy (static enthalpy).

(d) A volute casing to collect the fluid and to further convert the kinetic energy into pressure energy (static enthalpy).

Besides these, there are intercoolers, generally integrated with the casing, in a multistage compressor. The casing is usually made of cast iron and the impeller, of alloy (chrome-nickel) steels. The maximum stress is developed at the root of the blades.

The diffuser is normally vaneless type as it permits more efficient part load operation which is quite usual in any air-conditioning plant. A vaned diffuser will certainly cause shock losses if the compressor is run at reduced capacity and flow.

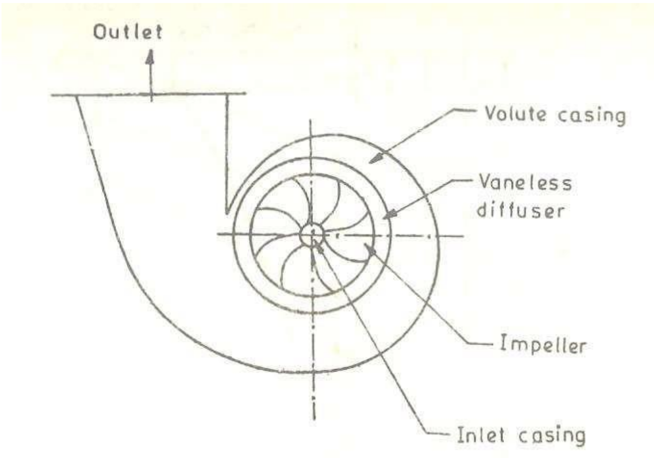



**Figure 3.4: Elements of a Centrifugal Compressor**

### Performance Characteristics

The principal performance curve of a centrifugal machine is the head-flow characteristic.

With $\gamma_2$ = radius of impeller

$\beta_2$ = angle of exit at impeller tip

C = velocity with suffix *r* for radial, *u* for tangential and 2 for exit

$\omega$ = angular speed of impeller in rad/s

u = velocity of impeller tip

we may write for the tangential velocity at the exit

$$C_{u2} = u_2 - C_{r2} \cot \beta_2 \qquad \ldots 3.14$$

We know that head developed with no pre-whirl is given by,

$$w = C_{u2} u_2 \qquad \ldots 3.15$$

form the above two equations we get,

$$w = u_2 (u_2 - C_{r2} \cot \beta_2)$$

$$= u_2^2 - u_2 C_{r2} \cot \beta_2$$

$$= (\omega r_2)^2 - (\omega r_2) C_{r2} \cot \beta_2 \qquad \ldots 3.16$$

Thus we find that for a given compressor, for which $\gamma_2$ and $\beta_2$ are fixed, and a rotating with certain speed, the head developed is a straight line function of the radial velocity $C_{r2}$. The flow rate a, in turn, is proportional to $C_{r2}$, The limiting head is $u_2^2$ which is developed at, $C_{r2}$=0, i.e., at zero flow rate. This occurs when the impeller is simply rotating in a mass of the fluid with the delivery valve closed.

It is seen that the nature of the characteristic depends on the outlet blade angle $\beta_2$ as follows:

Three types of blades are identified. They are backward-curved, radial and forward curved.

(a) For backward-curved blades, $\beta_2 < 90°$ head decreases with flow and hence with Q

(b) For radial blades, $\beta_2 = 90°$ head = $u_2^2$ = const.

(c) For forward-curved blades, $\beta_2 > 90°$, head increases with flow

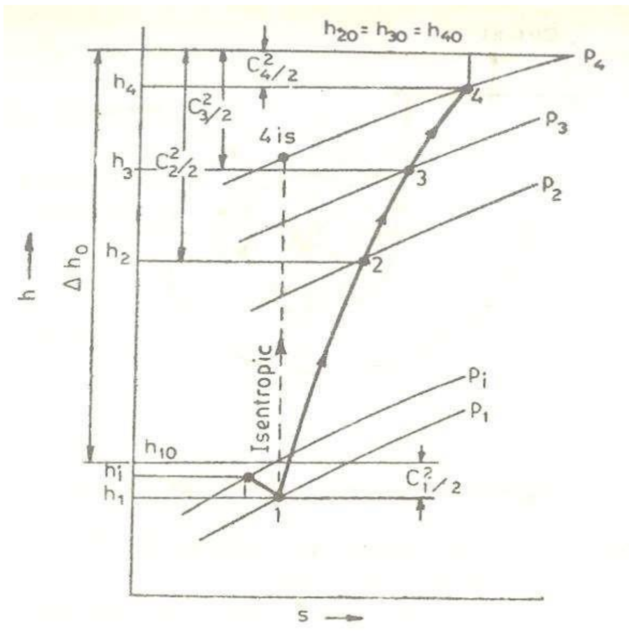



**Figure 3.5: Mollier Diagram of Centrifugal Stage**

From the point of view of optimal design, an outlet blade angle of 32° is normally preferred. A simple design will, however, have radial blades.

Figure 3.6 shows the theoretical head-flow characteristic for the three cases of angle $\beta_2$. For the case of backward-curved blades, it is a drooping characteristic.

The actual characteristic can, however, be obtained by considering the following losses as shown in Figure 3.6

(a) Leakage loss $L_1$ proportional to the head.

(b) Friction loss $L_2$ proportional to $\frac{C_{rel}^2}{2}$ and hence $Q^2$

(c) Entrance loss $L_3$ due to turning of the fluid to enter the impeller, being zero at the design point, which also corresponds to maximum efficiency.

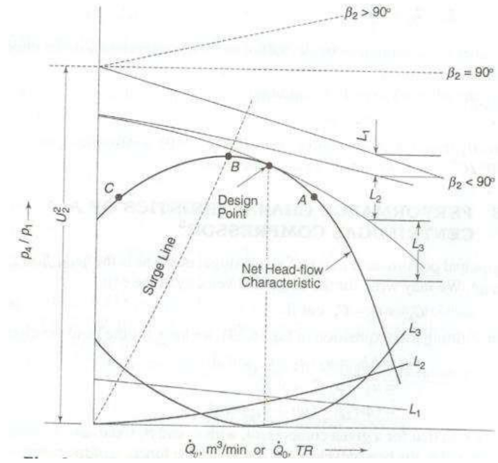



**Figure 3.6: Performance Characteristic and Losses of a Centrifugal Compressor**

## Surging

Consider A as the point of operation at full load. When the refrigeration load decreases, the point of operation shifts to the left until point B of maximum head is reached. If the load continues to decrease to the left of B, say to C, the pressure ratio developed by the compressor becomes less than the ratio required between the condenser and evaporator pressure. viz.,

$$\frac{p_4}{p_1} < \frac{p_k}{p_0}$$

Hence some gas flows back from the condenser to the evaporator, thus increasing the evaporator pressure and decreasing $\frac{p_k}{p_0}$ .The point of operation suddenly shifts to A. As the refrigeration load is still less, the cycle will repeat itself. This phenomenon of reversal of flow in centrifugal compressors is called surging. It occurs when the load decreases to below 35 per cent of the rated capacity and causes severe stress conditions in the compressor as a result of hunting.

## Capacity Control of Centrifugal Compressors

Centrifugal compressors require high tip speeds to develop the necessary pressure ratio. The high tip speed is achieved by employing either a large diameter impeller or high rpm or both. Because of large $u_2$, the velocities in general including the flow velocity C are high. Also, there must he a reasonable width of the shrouds to minimize friction and achieve high efficiency. Thus, because of the sufficiently large flow area (diameter D and width of shrouds b) required and large flow velocity, the satisfactory volume that can be handled by a centrifugal compressor is about 30-60 cubic metres per minute. A single centrifugal compressor, therefore, can be and industrial processing, cold storages and freezing, as high displacement. low stage or booster compressors at -90 to -100°C evaporator temperature with R-12, R-22 and ammonia. They are available in 10 to 600 hp sizes with 2 to 120 cubic metres per minute displacement in one unit.

designed for a minimum capacity approximately of the order of 250 TR with R 11 and 150 TR with R 113 for the purpose of air conditioning. One of the methods to control the capacity of the compressors is by varying the compressor speed through a speed-reduction gear. The decrease in speed results in an operation on a lower head-flow characteristic giving a lower volume flow rate corresponding to the same pressure ratio.

Capacity can be controlled by the use of variable inlet whirl vanes that are frequently employed with a constant speed drive. The capacity is varied by changing the angle at which the gas enters the impeller. The gas then enters with pre-rotation and this result in a decrease in flow.

## Rotary Compressor

Rotary compressors are positive displacement, direct-drive machines. There are essentially two designs of this compressor:

(a) Rolling piston type

(b) Rotating vane type

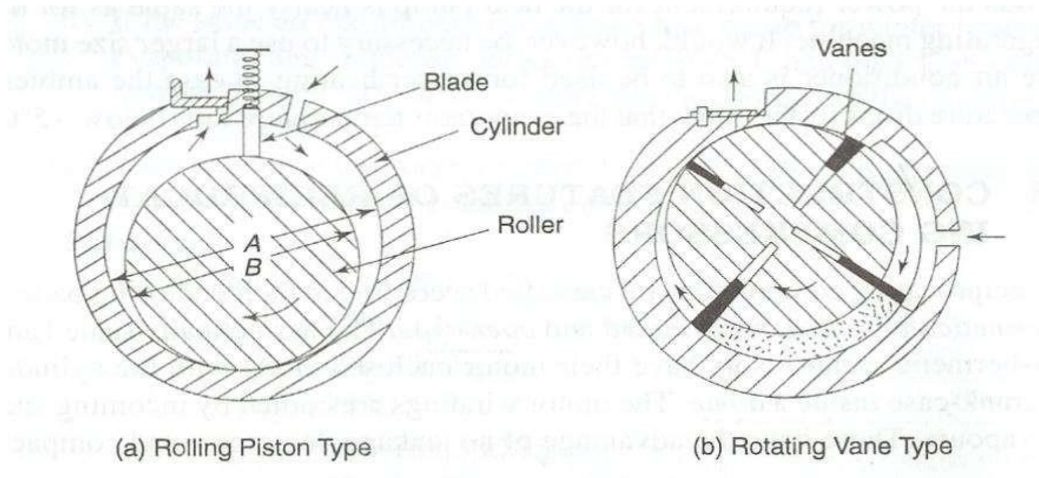



**Figure 3.7: Rotary Compressor**

In the rolling piston type, shown in Figure 3.7(a) the roller is mounted on an eccentric shaft with a single blade, which is always in contact with the roller by means of a spring. The theoretical piston displacement is

$$V_p = \frac{H(A^2 - B^2)}{4} \qquad \ldots 3.17$$

where A and B are respectively the diameters of the cylinder and rolling piston and H the lenght of the cylinder.

In the rotating vane type, as shown in Figure 3.7(b) with four vanes, the rotor is concentric with the shaft. The vanes slide within the rotor but keep contact with the cylinder. The assembly of rotor and the vanes is off-centre with respect to the cylinder.

In both designs, the whole assembly is enclosed in a housing (not shown in the figures), filled with oil and remains submerged in oil. An oil film forms the seal between the high-pressure and the low-pressure sides. When the compressor stops, this seal is lost and the pressure equalizes.

Rotary compressors have high volumetric efficiencies due to negligible clearance. They are normally used in a single stage up to a capacity of 5 TR with R-114. Large rotary compressors are used in low-temperature fields, such as in chemical

# CONDENSERS

The functions of the condenser are to desuperheat the high pressure gas, condense it and also sub- cool the liquid.

Heat from the hot refrigerant gas is rejected in the condenser to the condensing medium-air or water. Air and water are chosen because they are naturally available. Their normal temperature range is satisfactory for condensing refrigerants.

Like the evaporator, the condenser is also heat-exchange equipment.

## Types of Condenser

There are three types of condensers, viz.

(a) Air- cooled,

(b) Water-cooled and

(c) Evaporative.

As their names imply, air-cooled condensers use air as the cooling medium, water-cooled condensers use water as the medium and the evaporative condenser is a combination of the above, i.e. uses both water and air.

## Air-Cooled Condensers

There are two types under this category, viz. (a) natural convection and (b) forced-air type.

### Natural Convection Condenser

Air movement over the surface of condenser tubes is by natural convection. As air comes in contact with the warm-condenser tubes, it absorbs heat from the refrigerant and thus the temperature of the air increases. Warm air being lighter, rises up and in its place cooler air from below rises to take away the heat from the condenser. This cycle goes on. Since air moves very slowly by natural convection, the rate of flow of heat from the refrigerant to air will be small. Thus a natural convection condenser is not capable of rejecting heat rapidly. Therefore a relatively large surface area of the condenser is required. Hence the use of this type of condenser is limited to very small units such as domestic refrigerators. It, however, requires very little maintenance.

In the small units, the condenser is fixed at the rear of the refrigerator cabinets. Generally, steel tubes are used, steel being cheaper than copper. To increase the heat-transfer area, wires are welded to the condenser tubes. These wires provide mechanical strength to the coil as well. In certain designs, widely-spaced fins are used. It is necessary to space the fins quite widely to avoid resistance to free (natural convection) air movement over the condenser.

Still another design is the plate-type. The condenser coil is fastened to a plate. The plate being in contact with the condenser tubes, the surface area of the condenser is increased. The plate-type condenser is mounted on the back of the refrigerator cabinet with a small gap between the cabinet and the plate. This gap gives an air- flue effect and facilitates better natural convection air currents.

It is obvious that while locating refrigerators or deep-freezes cabinets with a natural convection condenser fixed on the cabinet, sufficient care should be taken to allow free air movement. Also they should not be near an oven or any warm location.

**Forced-air Circulation Condenser**

This type employs a fan or blower to move air over the condenser coil at a certain velocity. The condenser coil is of the finned type. Fins in such coils are closely spaced (ranging between 8 and 17 fins per inch). The space between the fins gets choked with dirt and lint. Therefore to obtain optimum capacity, the fins should be kept clean. For circulating air over the condenser, fans are mounted on the shaft/pulley of the compressor motor. For bigger-capacity plants a separate motor is used to drive the fan or blower as also for hermetic-compressor units.

### 3.3.3 Water Cooled Condensers

There are three types of condensers which fall under this category:

(a) tube-in-tube or double pipe,

(b) shell-and-coil, and

(c) shell-and-tube.

**Tube-in-Tube or Double Pipe Condenser**

In this type, a smaller diameter pipe inserted inside a bigger diameter pipe is bent to the desired form. Water flows through the inner tube and the refrigerant through the annular space between the two tubes; the flow of refrigerant and water being arranged in opposite direction to get the maximum benefit of heat-transfer. Due to the impurities present in water, scale can form on the water-side of the tube which can impede the heat transfer; also muck can settle on the surface. Therefore it becomes necessary to periodically clean the water tube. But in the tube-in-tube system, cleaning is not easy, unless a removable header is provided to connect all the tubes.

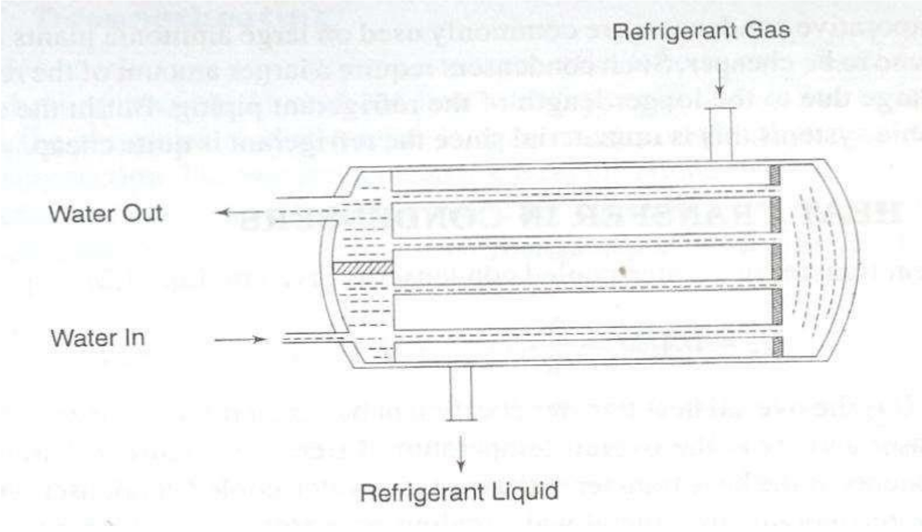



**Figure 3.9: Schematic Representation of a Two-Pass Water-Cooled Shell and Tube Condenser**

### Shell-and-Coil Condenser

It consists of a welded-steel shell containing a coil of finned tubing. Water flows in the coil, the refrigerant being in the shell. Since the tube bundle is in the form of a coil, the water-side of the tube cannot be brushed but can only be cleaned chemically.

### Shell-and-Tube Condenser

Figure 3.9 shows a typical shell-and-tube condenser. This is similar in construction to the flooded chiller. A number of straight tubes with integral fins are stacked inside a cylindrical shell, the tube ends expanded into tube sheets which are welded to the shell at both the ends. Intermediate tube supports are provided in the shell to avoid sagging and rattling of the tubes. Since it is very easy to clean the water-side and also, it can be easily repaired, this type of water-cooled condenser is very popular. Since ammonia affects copper, steel tubes are used for ammonia condensers. Water flows through the condenser water tubes while the refrigerant remains in the shell.

Since copper has a high thermal expansion and contraction rate, the tube tends to move back and forth in the tube sheets due to the variations in temperature.

To prevent the tubes from getting loose at the rolled ends due to this action, the holes in the tube sheets have small grooves. They are only a few hundredths of mm deep. When the tube ends are rolled or expanded in the tube-sheet holes, the copper tubes also expand into the grooves, thereby effectively anchoring the tube ends to the tube sheets and preventing movement of the tubes at the ends. However the expansion forces can cause the tubes to bow.

Removable water boxes are provided at the ends of the condenser to facilitate brushing of the water tubes.

Hot (superheated) refrigerant gas enters at the top of the shell and gets cooled (desuperheated) and condensed as it comes in contact with the water tubes. The condensed liquid drains off to the bottom of the shell. In some condensers extra rows of water tubes are provided at the lower end of the condenser for sub-cooling the liquid below the condensing temperature.

Often the bottom portion of the condenser also serves as the receiver, thereby eliminating the necessity of a separate receiver. However, if the maximum storage capacity (for the refrigerant) of the condenser is less than the total charge of the system, a receiver of adequate capacity has to be added in case the pump down facility is to be provided-such as in ice-plants, cold-storage jobs, etc.

Care should be taken not to overcharge the system with the refrigerant. This is because an excessive accumulation of liquid in the condenser tends to cover too much of the water tubes and reduce the heat-transfer surface available for condensing the high-pressure gas. This result in increasing the head pressure and condensing temperature, and excessive overcharge can create hydraulic pressures.

A fusible plug or safety pressure relief valve is fixed on the shell of the condenser to protect the high side of the refrigeration system against excessive pressures.

## Evaporative Condenser

These condensers (Figure 3.10) have some features of both air-and water-cooled types. Both air and water are employed as a condensing medium. Water is pumped from the sump of the evaporative condenser to a spray header and sprayed over the condenser coil. At the same time a fan thaws air from the bottom-side of the condenser and discharges it out at the top of the condenser. An eliminator is provided above the spray header to stop particles of water from escaping along with the discharge air. The spray water coming in contact with the condenser tube surface evaporates into the air stream. The source of heat for vaporizing the water is taken from the refrigerant, thereby condensing the gas.

The evaporative condenser combines the functions of the water-cooled condenser and the cooling tower and hence occupies less space. Moreover, it needs less power than a water-cooled condenser. But the most troublesome point about the evaporative condenser is the difficulty in keeping the surface of the condenser coil clean. The condenser coil being both hot and wet in operation, the dirt carried along with the air stream forms a hard layer on the condenser. Scale also forms a hard layer if hard water is used. Once these hard layers are allowed to form, it is never possible to effectively clean the coil. So the capacity of the condenser gets substantially affected. Because of this maintenance problem, evaporative condensers are not much in favour.

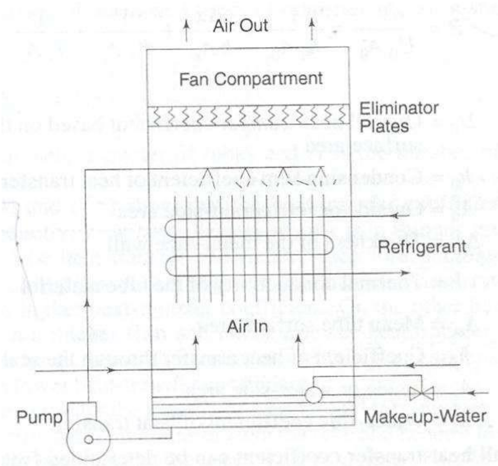



**Figure 3.10: Evaporative Condenser**

**Heat Transfer in Condensers** The heat transfer in a water-cooled condenser is described by the Equation given below

$$\dot{Q} = UA\Delta t = \frac{\Delta t}{R} \qquad \ldots 3.18$$

where U is the overall heat transfer coefficient based on the surface area A of the condenser and Δt is the overall temperature difference. Figure 3.11 shows the components of the heat-transfer resistance in a water-cooled condenser, viz., the outside refrigerant film, metal wall, scaling on water-side surface and inside-water film. The overall resistance is obtained by adding all the resistances which are in series.

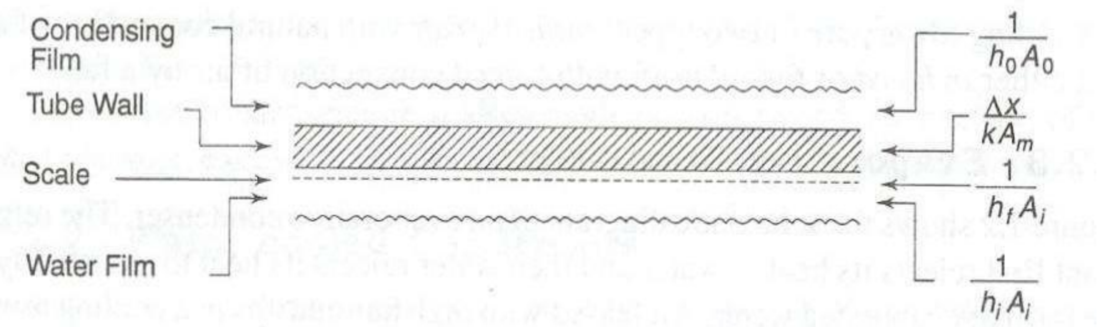



**Figure 3.11: Thermal Resistance in Water-Cooled Condenser**

$$R = \frac{1}{U_0 A_0} = \frac{1}{h_0 A_0} + \frac{\Delta x}{k A_m} + \frac{1}{h_f A_f} + \frac{1}{h_i A_i} \qquad \ldots 3.19$$

where

$U_0$ = Overall heat-transfer coefficient based on the outside surface area

$h_0$= Condensing film coefficient of heat transfer

$A_0$ = Outside or refrigerant-side area

k = Thermal conductivity of the tube material

$A_m$ = Mean tube surface area

$h_f$= Coefficient of heat transfer through the scale

$A_i$ = Inside or water-side area

$h_i$= Water-side coefficient of heat transfer.

Thus the overall heat-transfer coefficient can be determined from the above Equation 3.19 after estimating the individual resistances.

## EVAPORATORS

The process of heat removal from the substance to be cooled or refrigerated is done in the evaporator. The liquid refrigerant is vaporized inside the evaporator (coil or shell) in order to remove heat from a fluid such as air, water etc.

Evaporators are manufactured in different shapes, types and designs to suit a diverse nature of cooling requirements. Thus, we have a variety of types of evaporators, such as prime surface types, finned tube or extended surface type, shell and tube liquid chillers, etc.

# Types of Evaporator

Evaporators are classified into two general categories-the ‘dry expansion’ evaporator and ‘flooded’ evaporator.

### Dry Expansion Evaporator

In the dry-expansion evaporator, the liquid refrigerant is generally fed by an expansion valve. The expansion valve controls the rate of flow of refrigerant to the evaporator in such a way that all the liquid is vaporized and the vapour is also superheated to a limited extent by the time it reaches the outlet end. At the inlet of the evaporator, the

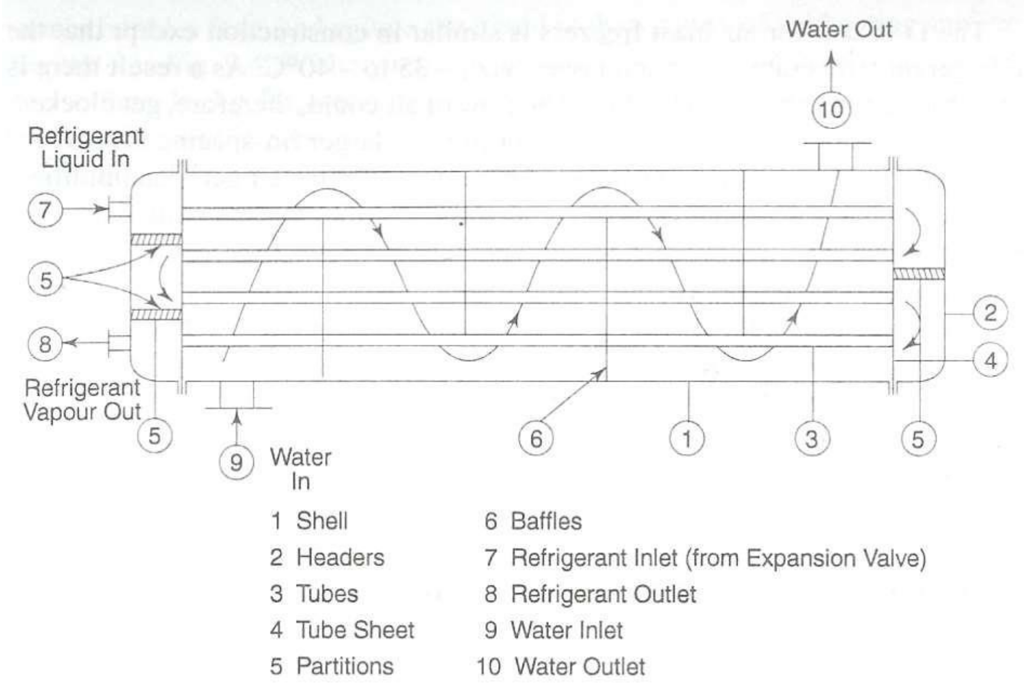



**Figure 3.12: Direct Expansion Evaporator**

refrigerant is predominantly in the liquid form with a small amount of vapour formed as a result of flashing at the expansion valve. As the refrigerant passes through the evaporator, more and more liquid is vaporized by the load. The refrigerant, by the time it reaches the end of the evaporator, is purely in the vapour state and that too superheated. Thus the evaporator in its length is filled with a varying proportion of liquid and vapour. The amount of liquid in the evaporator will vary with the load on the evaporator. The inside of the evaporator is far from ‘dry’ but wetted with liquid. All the same, this type is called the ‘dry-expansion’ system to distinguish it from the ‘flooded’ system and also probably because by the time the refrigerant reaches the evaporator outlet it is no more wet (no liquid) but dry (superheated) vapour.

### Flooded Evaporator

In a flooded-type evaporator a constant refrigerant liquid level is maintained. A float valve is used as the throttling device which maintains a constant liquid level in the evaporator. Due to the heat supplied by the substance to be cooled, the liquid refrigerant vaporizes and so the liquid level falls. The float valve opens to admit more liquid and thus maintains a constant liquid level. As a result, the evaporator is always filled with liquid to a level as determined by the float adjustment and the inside surface is wetted with liquid. Thus this type is called the flooded evaporator. The heat-transfer efficiency increases because the entire surface is in contact with the liquid refrigerant and, therefore, the flooded evaporator is more efficient. But the refrigerant charge is relatively large as compared to the dry-expansion type. As the evaporator is filled with liquid, it is obvious that the vapour from the evaporator will not be superheated but will be at saturation. To prevent liquid carry over to the compressor, accumulators' are generally used in conjunction with flooded evaporators. The accumulator also serves as the chamber for the liquid level float valve. The evaporator coil is connected to the accumulator and the liquid flow from the accumulator to the evaporator coil is generally by gravity. The vapour formed by the vaporization of the liquid in the coil being lighter, rises up and passes on to the top of the accumulator from where it enters the suction line as shown in Figure 3.13. In some cases, liquid eliminators are provided in the accumulator top to prevent the possible carry-over of liquid particles from the accumulator to the suction line. Further, a liquid-suction heat exchanger is used on the suction line to superheat the suction vapour. For some applications, a refrigerant liquid pump is employed for circulating the liquid from the accumulator to the evaporator coil and such a system is called a 'liquid-overfeed system'.

While the terms 'dry expansion' and 'flooded' indicate the manner in which the liquid refrigerant is fed into the evaporator and circulated, the terms 'natural convection' and 'forced convection' describe the way in which the fluid (air or liquid) is cooled/circulated around the evaporator.

Natural convection relies on the movement in a fluid, where the colder layer at the top being heavier falls down and the warmer layer rises up. By keeping an evaporator in the topmost portion of an insulated cabin, the air inside the cabin gets cooled by natural convection. A domestic refrigerator is a typical example. In 'forced-convection' types, the fluid is 'forced' over the evaporator by means of a fan or a liquid pump. In a room air conditioner, a fan continuously circulates the room air over the cooling coil and thus cools the room air. In a chilled-water system, a water pump or brine pump circulates the fluid through the chiller and cooling coils. For a 'coil-in-tank' arrangement, such as in an ice plant, an agitator is used to move the brine over the cooling coil with a certain amount of velocity.

### 3.4.2 Heat Transfer in Evaporators

The three heat-transfer resistances in evaporators are:

(a) Refrigerant side for the transfer of heat from solid surface to the liquid refrigerant.

(b) Metal wall.

(c) Cooled-medium side which could be due to air, water, brine or any other fluid or a wetted surface on a cooling and dehumidifying coil.

The heat transfer from solid surface to the evaporating refrigerant is of primary interest here. However, the mechanism of boiling i so complex because of the influence of such factors as surface tension, saturation temperature, latent heat and nature of the solid surface, in addition to the usual transport properties, that it is very difficult to predict the heat-transfer coefficient analytically. Nevertheless, no attempt is made here to present correlations applicable to evaporating refrigerants which are available in the large amount of published information available on the subject.

In commercial equipment, the boiling process occurs in two types of situations: one, of pool boiling as in flooded evaporators with refrigerant boiling the shell-side and the other, of flow or forced convection boiling as in direct- expansion evaporators with refrigerant on the tube-side.

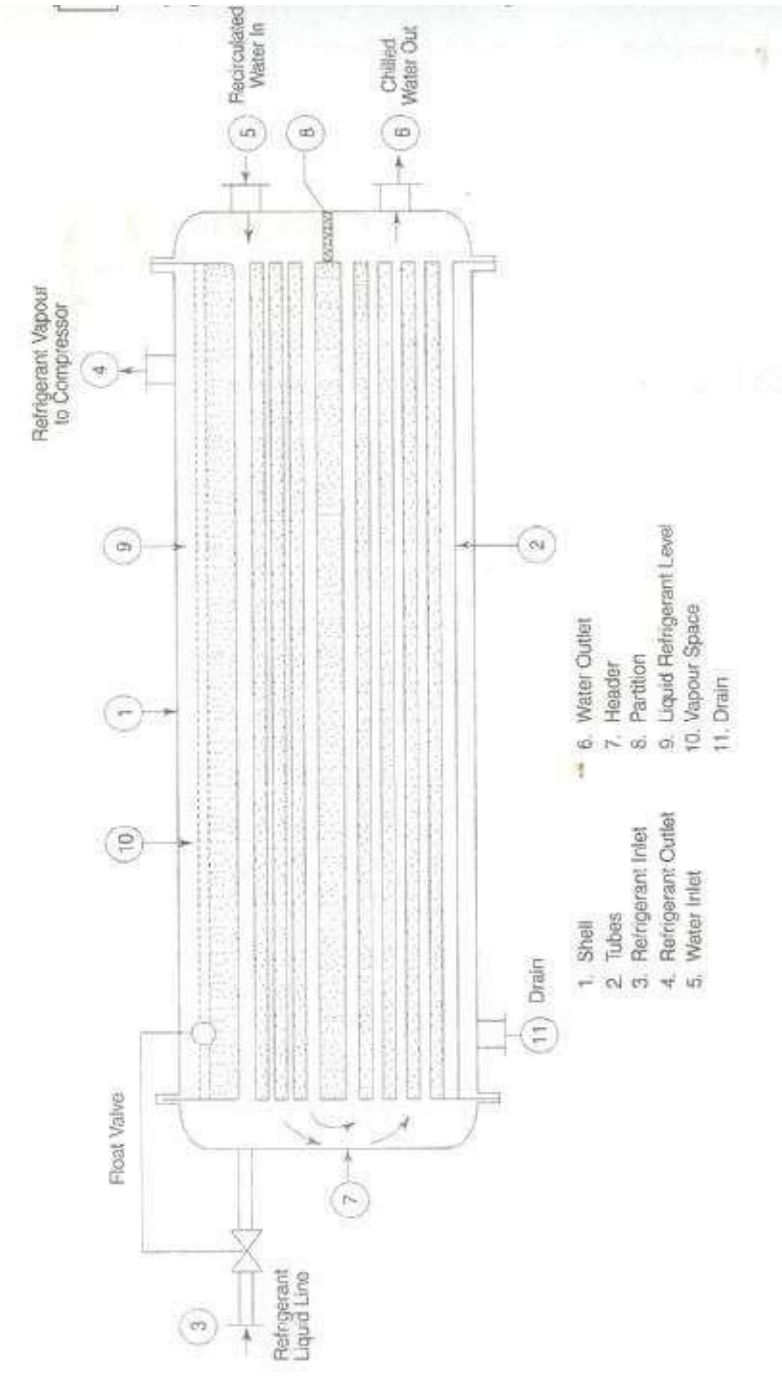



**Figure 3.13: Flooded Evaporator**

## CHAPTER 5 REFRIGERANT FLOW CONTROL,REFRIGERANT & APPLICATION OF REFRIGERANT

## EXPANSION DEVICES

There are different types of expansion or throttling devices. The most commonly used are:

(a) Capillary tube,

(b) Float valves,

(c) Thermostatic expansion valve.

### Capillary Tube

Instead of an orifice, a length of a small diameter tube can offer the same restrictive effect. A small diameter tubing is called ‘capillary tube’, meaning ‘hair-like’. The inside diameter of the capillary used in refrigeration is generally about 0.5 to 2.28 mm (0.020 to 0.090’). The longer the capillary tube and/or the smaller the inside diameter of the tube, greater is the pressure drop it can create in the refrigerant flow; or in other words, greater will be the pressure difference needed between the high side and low side to establish a given flow rate of the refrigerant.

The length of the capillary tube of a particular diameter required for an application is first roughly determined by empirical calculations. It is then further correctly established by experiments. The capillary tube is not self-adjusting. If the conditions change, such as an increase in the discharge/condenser pressure due to a rise in the ambient temperature, reduction in evaporator pressure, etc. the refrigerant flow-rate will also change. Therefore a capillary tube, selected for a particular set of conditions and load will operate somewhat less efficiently at other conditions. However if properly selected, the capillary tube can work satisfactorily over a reasonable range of conditions.

As soon as the plant stops, the high and low sides equalize through the capillary tube. For this reason, the refrigerant charge in a capillary tube system is critical and hence no receiver is used. If the refrigerant charge is more than the minimum needed for the system, the discharge pressure will go up while in operation. This can even lead to the overloading of the compressor motor. Further, during the off-cycle of the unit, the excess amount will enter the cooling coil and this can cause liquid flood back to the compressor at the time of starting. Therefore, the refrigerant charge of the capillary tube system is critical. For this reason, a refrigerant liquid receiver cannot be used. The charge should be exactly the quantity as indicated by the manufacturer of the refrigeration unit.

Since the capillary tube equalizes the high side with the low side during the off-cycle, the idle pressures at the discharge and suction of the compressor will be equal. Therefore at the time of starting, the compressor motor need not overcome the stress of the difference of pressure in the suction and the discharge sides. In other words the compressor is said to start unloaded. This is a great advantage as a low starting torque motor is sufficient for driving the compressor.

The capillary tube is quite a simple device and is also not costly. Its pressure equalization property allows the use of a low starting torque motor. The liquid receiver is also eliminated in a capillary tube system because of the need to limit the refrigerant charge. All these factors help to reduce the cost of manufacture of the systems employing a capillary tube as the throttling device.

the tank. As the water level rises, the float ball (which is hollow) floats on the water and gradually rises according to the water level, throttling the water through the valve. Ultimately when the tank is full, the float valve completely closes the water supply. As the water from the tank is used, the water level falls down; the float ball also lowers down, opening the valve according to the level of water in the tank.

The low-side float valve also acts in the same way in a refrigeration system. As the name implies the float valve is located in the low pressure side of the system. It is fixed in a chamber (float chamber) which is connected to the evaporator. The valve assembly consists of a hollow ball, a float arm, needle valve and seat. The needle valve-seat combination provides the throttling effect similar to the expansion valve needle and seat. The movement of the float ball is transmitted to the needle valve by the float arm. The float ball being hollow floats on the liquid refrigerant. The needle valve and seat are located at the inlet of the float chamber. As the liquid refrigerant vaporizes in the evaporator, its level falls down in the chamber. This causes the float ball to drop and pull the needle away from the seat, thereby allowing enough liquid refrigerant to flow into the chamber of the evaporator to make up for the amount of vaporization. When enough liquid enters, the float ball rises and ultimately closes the needle valve when the desired liquid level is reached. The rate of vaporization of liquid and consequent drop in the level of the liquid in the evaporator is dependent on the load. Thus the movement of the float ball and amount of opening of the float valve is according to the load on the evaporator. The float valve responds to liquid level changes only and acts to maintain a constant liquid level in the evaporator under any load without regard for the evaporator pressure and temperature.

Like in the expansion valve, the capacity of the low-side float valve depends on the pressure difference across the orifice as well as the size of the orifice.

Low-side float valves are used for evaporators of the flooded-type system. In bigger capacity plants a small low-side float valve is used to pilot a liquid feed (and throttling) valve. According to the liquid level in the evaporator, the float valve transmits pressure signals to the main liquid feed valve to increase or decrease the extent of its opening. Thus the low-side float valve in such a system is called a 'pilot' and the liquid-feed valve is known as the pilot-operated liquid-feed valve.

**High-side Float Valve**

The high-side valve like the low-pressure float valve, is a liquid level sensing device and maintains a constant liquid level in the chamber in which it is fixed. However it differs from the low-side float valve in the following respects.

(a) The high-side float valve and its chamber are located at the high-pressure side of the system, while the low-side float valve is located at the low-pressure side of the system.

(b) The needle and seat of the valve are at the outlet of the chamber as

The high-side float chamber is located between the condenser and evaporator. The liquid condensed in the condenser flows down to the float chamber.

(c) In the high-side float valve, the valve opens on a rise in the liquid level in the chamber, just the opposite action of the low-side float valve, which closes on a rise in liquid level in the chamber.
against the needle valve being at the inlet of the chamber in the low-side float.

As the liquid level rises in the chamber, the float ball also rises, thereby opening the needle valve. As the liquid level falls in the chamber, the float valve tends to close the seat orifice. It is obvious that refrigerant vapour is condensed in the condenser at the same rate at which the liquid vaporizes in the evaporator; the float chamber receives and feeds liquid to the evaporator at the same rate. Since the rate of vaporization of the liquid in the evaporator is according to the load, the high-side float obviously works as per the load.

This type of float valve is generally used in centrifugal-refrigeration plants.

Refrigerant feed/throttling devices for flooded chillers are usually the low-side or high-side float valve. For example, in centrifugal plants, the chiller is of the flooded type and generally high-side float valves are used as throttling devices. In a flooded chiller working in conjunction with a reciprocating compressor, a low-side float valve is used as the throttling and refrigerant liquid flow control.

## Thermo - static Expansion Valve

The name ‘thermostatic-expansion valve’ may give the impression that it is a temperature control device. It is not a temperature control device and it cannot be adjusted and used to vary evaporator temperature. Actually TEV is a throttling device which works automatically, maintaining proper and correct liquid flow as per the dictates of the load on the evaporator. Because of its adaptability to any type of dry expansion application, automatic operation, high efficiency and ability to prevent liquid flood backs, this valve is extensively used.

The functions of the thermostatic-expansion valve are:

(a) To reduce the pressure of the liquid from the condenser pressure to evaporator pressure,

(b) To keep the evaporator fully active and

(c) To modulate the flow of liquid to the evaporator according to the load requirements of the evaporator so as to prevent flood back of liquid refrigerant to the compressor.

It does the last two functions by maintaining a constant superheat of the refrigerant at the outlet of the evaporator. It would be more appropriate to call it a ‘constant superheat valve’.

The important parts of the valve are:

Power element with a feeler bulb, valve seat and needle, and a superheat adjustment spring.

# REFRIGERANTS

## FUNCTION OF REFRIGERANT

The refrigerant is heat carrying mediums which during their cycle (i.e. Compression, condensation, expansion and evaporation) in the refrigeration system absorb heat from a low temperature system and discard the heat so absorbed to a higher temperature system. The natural ice and a mixture of ice and salt were the first refrigerants. In 1834, ether, ammonia, sulphur dioxide, methyl chloride and carbon dioxide came into use as refrigerants in compression cycle refrigeration machines. The suitability of refrigerant for a certain application is determined by its physical, thermodynamic, chemical properties and by various practical factors. There is no one refrigerant which can be used for all types of applications i.e. there is no ideal refrigerant.

## CLASSIFICATION OF REFRIGERANTS

1. Primary Refrigerants
2. Secondary refrigerants

The refrigerants which directly take parting the refrigeration system are called primary refrigerants whereas the refrigerants which are first cooled by primary refrigerants and then used for cooling purposes are known as secondary refrigerants.

The primary refrigerants are further classified into the following four groups:

1. Halo carbon refrigerants
2. Azeotrope refrigerants
3. Inorganic refrigerants
4. Hydro-carbon refrigerants

## 1. Halo-carbon Refrigerants

The American society of Heating, Refrigeration and Air-conditioning Engineers (ASHRAE) identifies 42 halo-carbon compounds as refrigerants, but only a few of them are commonly used. The following table gives some of the commonly used halo-carbon refrigerants:

**Table 1: Halo carbon refrigerants**

| *Refrigerant number* | *Chemical name* | *Chemical formula* |
|---|---|---|
| R−11 | Trichloromonofluoromethane | $CCl_3F$ |
| R-12 | Dichlorodifluoromethane | $CCl_2F_2$ |
| R−13 | Monochlorotrifluoromethane | $CClF_3$ |
| R−14 | Carbontetrafluoride | $CF_4$ |
| R−21 | Dichloromonofluoromethane | $CHCl_2F$ |
| R−22 | Monochlorodifluoromethane | $CHClF_2$ |
| R−30 | Methylene chloride | $CH_2Cl_2$ |
| R−40 | Methyl chloride | $CH_3Cl$ |
| R−100 | Ethyl chloride | $C_2H_5Cl$ |
| R−113 | Trichlorotrifluoroethane | $CCl_2FCClF_2$ |
| R−114 | Dichlorotetrafluoroethane | $CClF_2CClF_2$ |
| R-115 | Monochloropentafluoroethane | $CClF_2CF_3$ |

## 2. Azeotropes:

An azeotrope is a suitable mixture of two Refrigerants which cannot be separated into its compounds by distillation. An azeotrope evaporates and condenses as a single substance that the vapours have the same composition as the liquid.

**Table 2: Azeotrop**

| Refrigerant number | Azeotrpic mixing refrigerants | Chemical formula |
|---|---|---|
| R-500 | 73.8% R-12 and 26.2% R-152 | $CCl_2F_2/CH_3CHF_2$ |
| R-502 | 48.8% R-22 and 51.2% R-115 | $CHClF_2/CClF_2CF_3$ |
| R-503 | 40.1% R-23 and 59.9% R-13 | $CHF_3/CClF_3$ |
| R-504 | 48.2% R-32 and 51.8% R-115 | $CH_2F_2/CClF_2CF_2$ |

### 3. Hydrocarbons:

These are organic compounds. These have a high tendency to burn or to form combustible mixtures with a wide range of concentration of air though these possess satisfactory thermodynamic properties. These are now not used much in common applications but are found to be suitable as refrigerants in petroleum and petro-chemical industries.

**Table 3: Hydrocarbons**

| Refrigerant number | Chemical name | Chemical formula |
|---|---|---|
| R-170 | Ethane | $C_2H_6$ |
| R-290 | Propane | $C_3H_3$ |
| R-600 | Butane | $C_4H_{10}$ |

**4. Unsaturated Organic Compounds:**

These are hydrocarbons with mainly ethylene or propylene base.

**Table 4: Unsaturated Organic Compounds**

| | | |
|---|---|---|
| R-1120 | Trichloroethylene | $C_2H_2Cl_3$ |
| R-1130 | Dichloroethylene | $C_2H_2Cl_2$ |
| R-1150 | Ethylene | $C_2H_4$ |
| R-1270 | Propylene | $C_3H_6$ |

**5. Inorganic Compounds:**

The inorganic compounds under this group were universally employed before the introduction of hydrocarbons compounds. Many of them are being still used in various refrigeration applications.

**Table 5: Inorganic Compound**

| Refrigerant number | Chemical name | Chemical formula |
|---|---|---|
| R-717 | Ammonia | $NH_3$ |
| R-729 | Air | ___ |
| R-744 | Carbon dioxide | $CO_2$ |
| R-764 | Sulphur dioxide | $SO_2$ |
| R-118 | Water | $H_2O$ |

## PROPERTIES OF REFRIGERANTS

**1. R-717 (Ammonia):** Ammonia is amongst the oldest of all the refrigerants and still used widely in the refrigeration applications. Ammonia refrigerant is commonly known as R717 and its chemical formula is $NH_3$. Its molecular weight is 17 and boiling point is -28 degree F (-2.22 degree C).

Different properties of ammonia are given below:

**2.** One of the biggest advantages of ammonia gas as the refrigerant is that it is safe to the environment and does not cause any depletion of the ozone layer.

**3.** The discharge temperature of the ammonia refrigerant from the compressor is high.

**4.** Ammonia is non-corrosive in nature, however, in presence of moisture it tends to become corrosive to copper, brass and other non-ferrous materials.

**5.** Ammonia refrigerant is non-miscible with oil so it does not mix with the oil in the crankcase of the compressor.

**6.** The leak testing of ammonia from the refrigeration system can be done either by using sulfur sticks or soap solution. When ammonia reacts with sulfur, a dense smoke is formed.

**2. R-22 (Freon):** R22 refrigerant is one of the most commonly used refrigerants in the air conditioning systems. R22 is the short name for the halocarbon compound $CHClF_2$ (Dichlorodifluoromethane), which is used as the refrigerant. R stands for the refrigerant. In the number "22" second "2" denotes the number of the fluorine atoms in the compound.

- R-22 was used extensively in domestic, commercial as well as industrial low-temperature systems to evaporator temperatures as low as -87°C.
- Both atmospheric pressure and discharge temperature are higher.
- Evaporator temperatures are between -28 to -40°C.
- The ability of R22 to absorb moisture is comparatively greater, therefore, less trouble due to ice formation.
- Fluorocarbon based refrigerants are safe.
- Use a halide torch for leak detection.
- R22 was completely phased out due to its high ozone depleting potential.

**3. R-134(a) (HCF):** Coming from the HFC (hydrofluorocarbons) family of refrigerants, R134a is also known as tetrafluoroethane ($CH_2FCF_3$) consisting of two carbon atoms, 2 hydrogen atoms and 4 fluorine atoms. The properties of R134a refrigerant gas are discussed below:

- R134a is no-toxic, non-flammable and non-corrosive.
- R134a has a boiling point of -15.34 degree Fahrenheit or -26.3 degree Celsius that makes it exist in gas form when exposed to environment. This is a desired property as the boiling point of a refrigerant should be below the target temperature.
- R134a has a high heat of vaporization.
- Its solubility in water is 0.11% by weight at 77degree Fahrenheit or 25 degree Celsius.
- R134a has zero Ozone layer depleting properties and hence became popular as an ideal replacement for dichlorodifluoromethane (R-12), which was known to have an adverse impact on the Ozone layer.
- This refrigerant has a Global Warming Potential (GWP) of 1300. GWP is a relative measure of the amount of heat trapped in the atmosphere by a greenhouse gas.

**4. $CO_2$ (R-744):** R-744 has been used in refrigeration for many years. Carbon dioxide offers a long-term solution suitable for many applications in refrigeration and heating, from domestic applications utilizing heat pumps to provide hot water and heating to commercial applications. R-744 has 10 noteworthy characteristics:

- Non-toxic
- Non-flammable
- Low triple point
- Low critical point
- High pressure

- High refrigeration volumetric capacity
- High heat transfer characteristics
- Inexpensive
- Readily available

5. **R-12 (CFC):** Prior to the environmental issues of ozone layer depletion and global warming, it was the most widely used refrigerant. R-12 was used primarily in small capacity refrigeration and cold storage applications. It is:

- Non-toxic
- Non-flammable
- NBP = -29.8 Deg.C
- Hfg at NBP=165.8 kJ/kg
- Tcr =112.04 Deg. C
- Cp/Cv = 1.126
- ODP = 1.0
- GWP = 7300

6. **R-502 ($CHCLF_2$):** It is one of the newer refrigerant. It has replaced R22 because of difficulties of high discharge temperature and poor oil return experienced with R22.The boiling point of R 502 is -46°C. Its various properties are:

- It is non toxic in nature.
- It is non flammable and non irritating.
- It is stable and non corrosive.
- It is suitable for obtaining medium and low temperature range.
- The leak may be detected by soap solution, halide torch detector.

## PROPERTIES OF IDEAL REFRIGERANT

We have discussed above that there is no ideal refrigerant. A refrigerant is said to be ideal if it has all of the following properties:

- Low Boiling point
- High critical temperature
- High latent heat of vaporization
- Low specific heat of liquid
- Low specific volume of vapour
- Non-corrosive to metal
- Non-flammable and non-explosive
- Non-toxic
- Low cost
- Easy to liquefy at moderate pressure and temperature
- Easy of locating leaks by odour or suitable indicator, and
- Mixes well with oil.

## SELECTION OF REFRIGERANT

While selecting a refrigerant for a particular application, the thermodynamic, physical, chemical and safe working properties etc. of the refrigerant should be properly weighed. The following points are of primary consideration:

1. Working Temperature and pressure range.
2. Toxicity, flammability and corrosiveness.
3. Space Limitations.
4. Oil Miscibility.
5. Temperature required in the evaporator, condenser and that of the cooling medium available.

# Short and long questions of ch-4 & 5

1.a.What is the function of compressor in Refrigerator?
  b.Classify refrigerant compressor.
  c.What should be the condition of refrigerant before compression ?
  d.What is Multistage compression?
  e.What is the use of Condenser in refrigeration system?
  f.What is heat rejection factor for condenser?
  g.Classify condenser used in refrigeration of system.
  h.Give any 2 difference between air-cooled condenser and water-cooled condenser.
  I.What is fouling in condenser?
 J.Classify evaporator.
h.What is primary refrigerant?
k.What is secondary refrigerant?

2.a.With Neat sketch explain different types of Condenser used in refrigeration system.

b.With Neat sketch explain different types of evaporator used in refrigeration system.

c.With Neat sketch explain different types of expansion valve used in refrigeration system.
d.what are the desirable properties of Ideal refrigerant?
e.Explain working of a reciprocating compressor.
f.Classify primary refrigerant & discuss their various properties.

# PSYCHOMETRICS & COMFORT AIR CONDITIONING SYSTEMS

*Atmospheric air* makes up the environment in almost every type of air conditioning system. Hence a thorough understanding of the properties of atmospheric air and the ability to analyze various processes involving air is fundamental to air conditioning design.

*Psychrometry* is the study of the properties of mixtures of air and water vapour.

Atmospheric air is a mixture of many gases plus water vapour and a number of pollutants (Fig.27.1). The amount of water vapour and pollutants vary from place to place. The concentration of water vapour and pollutants decrease with altitude, and above an altitude of about 10 km, atmospheric air consists of only dry air. The pollutants have to be filtered out before processing the air. Hence, what we process is essentially a mixture of various gases that constitute air and water vapour. This mixture is known as *moist air*.

### 27.2.1. Basic gas laws for moist air:

According to the *Gibbs-Dalton law* for a mixture of perfect gases, the total pressure exerted by the mixture is equal to the sum of partial pressures of the constituent gases. According to this law, for a homogeneous perfect gas mixture occupying a volume V and at temperature T, each constituent gas behaves as though the other gases are not present (i.e., there is no interaction between the gases). Each gas obeys perfect gas equation. Hence, the partial pressures exerted by each gas, $p_1, p_2, p_3$ … and the total pressure $p_t$ are given by:

$$p_1 = \frac{n_1 R_u T}{V};\ p_2 = \frac{n_2 R_u T}{V};\ p_3 = \frac{n_3 R_u T}{V}\ \ldots\ldots \qquad (27.1)$$
$$p_t = p_1 + p_2 + p_3 + \ldots\ldots$$

where $n_1, n_2, n_3, \ldots$ are the number of moles of gases 1,2,3,…

Applying this equation to moist air.

$$p = p_t = p_a + p_v \qquad (27.2)$$

| | | |
|---|---|---|
| where $p = p_t$ | = | total barometric pressure |
| $p_a$ | = | partial pressure of dry air |
| $p_v$ | = | partial pressure of water vapour |

### 27.2.2. Important psychrometric properties:

Dry bulb temperature (DBT) is the temperature of the moist air as measured by a standard thermometer or other temperature measuring instruments.

Saturated vapour pressure ($p_{sat}$) is the saturated partial pressure of water vapour at the dry bulb temperature. This is readily available in thermodynamic tables and charts. ASHRAE suggests the following regression equation for saturated vapour pressure of water, which is valid for 0 to 100°C.

*Relative humidity (Φ)* is defined as the ratio of the mole fraction of water vapour in moist air to mole fraction of water vapour in saturated air at the same temperature and pressure. Using perfect gas equation we can show that:

The *sling psychrometer* is widely used for measurements involving room air or other applications where the air velocity inside the room is small. The sling psychrometer consists of two thermometers mounted side by side and fitted in a frame with a handle for whirling the device through air. The required air circulation (≈ 3 to 5 m/s) over the sensing bulbs is obtained by whirling the psychrometer (≈ 300 RPM). Readings are taken when both the thermometers show steady-state readings.

In the *aspirated psychrometer*, the thermometers remain stationary, and a small fan, blower or syringe moves the air across the thermometer bulbs.

The function of the wick on the wet-bulb thermometer is to provide a thin film of water on the sensing bulb. To prevent errors, there should be a continuous film of water on the wick. The wicks made of cotton or cloth should be replaced frequently, and only distilled water should be used for wetting it. The wick should extend beyond the bulb by 1 or 2 cms to minimize the heat conduction effects along the stem.

## 16.2 Psychrometric Terms

Though there are many psychrometric terms, yet the following are important from the subject point of view :

**1.** ***Dry air.*** The pure dry air is a mixture of a number of gases such as nitrogen, oxygen, carbon dioxide, hydrogen, argon, neon, helium etc. But the nitrogen and oxygen have the major portion of the combination.

2. ***Moist air.*** It is a mixture of dry air and water vapour. The amount of water vapour present in the air depends upon the absolute pressure and temperature of the mixture.

3. ***Saturated air.*** It is a mixture of dry air and water vapour, when the air has diffused the maximum amount of water vapour into it. The water vapours, usually, occur in the form of superheated steam as an invisible gas. However, when the saturated air is cooled, the water vapour in the air starts condensing, and the same may be visible in the form of moist, fog or condensation on cold surfaces.

4. ***Degree of saturation.*** It is the ratio of actual mass of water vapour in a unit mass of dry air to the mass of water vapour in the same mass of dry air when it is saturated at the same temperature.

5. ***Humidity.*** It is the mass of water vapour present in 1 kg of dry air, and is generally expressed in terms of gram per kg of dry air (g / kg of dry air). It is also called *specific humidity or humidity ratio.*

6. ***Absolute humidity.*** It is the mass of water vapour present in 1 $m^3$ of dry air, and is generally expressed in terms of gram per cubic metre of dry air ($g/m^3$ of dry air). It is also expressed in terms of grains per cubic metre of dry air. Mathematically, one kg of water vapour is equal to 15 430 grains.

7. ***Relative humidity.*** It is the ratio of actual mass of water vapour in a given volume of moist air to the mass of water vapour in the same volume of saturated air at the same temperature and pressure. It is briefly written as RH.

8. ***Dry bulb temperature.*** It is the temperature of air recorded by a thermometer, when it is not affected by the moisture present in the air. The dry bulb temperature (briefly written as DBT) is generally denoted by $t_d$ or $t_{db}$.

9. ***Wet bulb temperature.*** It is the temperature of air recorded by a thermometer, when its bulb is surrounded by a wet cloth exposed to the air. Such a thermometer is called *wet bulb thermometer. The wet bulb temperature (briefly written as WBT) is generally denoted by $t_w$ or $t_{wb}$.

10. ***Wet bulb depression.*** It is the difference between dry bulb temperature and wet bulb temperature at any point. The wet bulb depression indicates relative humidity of the air.

11. ***Dew point temperature.*** It is the temperature of air recorded by a thermometer, when the moisture (water vapour) present in it begins to condense. In other words, the dew point temperature is the saturation temperature ($t_{sat}$) corresponding to the partial pressure of water vapour ($p_v$). It is, usually, denoted by $t_{dp}$. Since $p_v$ is very small, therefore the saturation temperature by water vapour at $p_v$ is also low (less than the atmospheric or dry bulb temperature). Thus the water vapour in air exists in the superheated state and the moist air containing moisture in such a form (*i.e.* superheated state ) is said to be *unsaturated air*. This condition is shown by point *A* on temperature-entropy (*T-s*) diagram as shown in Fig. 16.1. When the partial pressure of water vapour ($p_v$) is equal to the saturation pressure ($p_s$), the water vapour is in dry condition and the air will be *saturated air*.

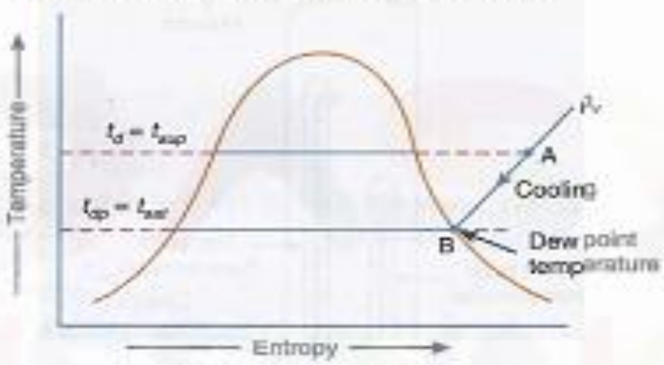



Fig. 16.1. *T-s* diagram.

If a sample of unsaturated air, containing superheated water vapour, is cooled at constant pressure, the partial pressure ($p_v$) of each constituent remains constant until the water vapour reaches the saturated state as shown by point *B* in Fig. 16.1. At this point *B*, the first drop of dew will be formed and hence the temperature at point *B* is called *dew point temperature*. Further cooling will cause condensation of water vapour.

12. ***Dew point depression.*** It is the difference between the dry bulb temperature and dew point temperature of air.

13. ***Psychrometer.*** There are many types of psychrometers, but the sling psychrometer, as shown in Fig. 16.2, is widely used. It consists of a dry bulb thermometer and a wet bulb thermometer mounted side by side in a protective case that is attached to a handle by a swivel connection so that the case can be easily rotated. The dry bulb thermometer is directly exposed to air and measures the actual temperature of the air. The bulb of the wet bulb thermometer is covered by a wick thoroughly wetted by distilled water. The temperature measured by this wick covered bulb of a thermometer is the temperature of liquid water in the wick and is called wet bulb temperature.

The sling psychrometer is rotated in the air for approximately one minute after which the readings from both the thermometers are taken. This process is repeated several times to assure that the lowest possible wet bulb temperature is recorded.

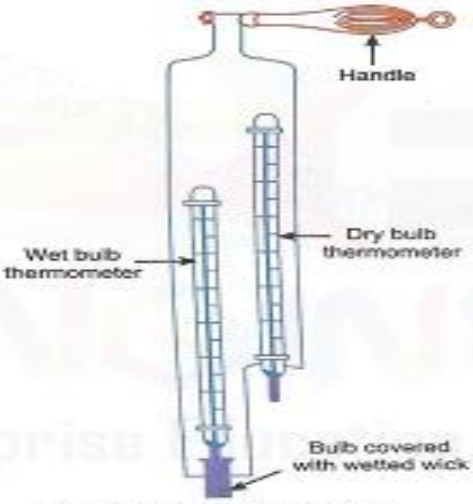



Fig. 16.2. Sling psychrometer.

Digital psychrometer.

## 16.3 Dalton's Law of Partial Pressures

It states, *"The total pressure exerted by the mixture of air and water vapour is equal to the sum of the pressures, which each constituent would exert, if it occupied the same space by itself."* In other words, the total pressure exerted by air and water vapour mixture is equal to the barometric pressure. Mathematically, barometric pressure of the mixture,

$$p_b = p_a + p_v$$

where $p_a$ = Partial pressure of dry air, and

$p_v$ = Partial pressure of water vapour.

## 16.4 Psychrometric Relations

1. ***Specific humidity, humidity ratio or moisture content.*** It is the mass of water vapour present in 1 kg of dry air (in the air-vapour mixture) and is generally expressed in g/kg of dry air. It may also be defined as the ratio of mass of water vapour to the mass of dry air in a given volume of the air-vapour mixture.

Let $p_a$, $v_a$, $T_a$, $m_a$ and $R_a$ = Pressure, volume, absolute temperature, mass and gas constant respectively for dry air, and

$p_v$, $v_v$, $T_v$, $m_v$ and $R_v$ = Corresponding values for the water vapour.

Assuming that the dry air and water vapour behave as perfect gases, we have for dry air,

$$p_a v_a = m_a R_a T_a \quad \text{... (i)}$$

and for water vapour,

$$p_v v_v = m_v R_v T_v \quad \text{... (ii)}$$

Also

$$v_a = v_v$$

and

$$T_a = T_v = T_d \quad \text{... (where } T_d \text{ is dry bulb temperature)}$$

From equations (*i*) and (*ii*), we have

$$\frac{p_v}{p_a} = \frac{m_v R_v}{m_a R_a}$$

$\therefore$ Humidity ratio, $$W = \frac{m_v}{m_a} = \frac{R_a\, p_v}{R_v\, p_a}$$

Substituting $R_a = 0.287$ kJ/kg K for dry air and $R_v = 0.461$ kJ/kg K for water vapour in the above equation, we have

$$W = \frac{0.287 \times p_v}{0.461 \times p_a} = 0.622 \times \frac{p_v}{p_a} = 0.622 \times \frac{p_v}{p_b - p_v}$$
$$\ldots (\because\ p_b = p_a + p_v)$$

Consider unsaturated air containing superheated vapour at dry bulb temperature $t_d$ and partial pressure $p_v$ as shown by point $A$ on the $T$-$s$ diagram in Fig. 16.3. If water is added into this unsaturated air, the water will evaporate which will increase the moisture content (specific humidity) of the air and the partial pressure $p_v$ increases. This will continue until the water vapour becomes saturated at that temperature, as shown by point $C$ in Fig. 16.3, and there will be more evaporation of water. The partial pressure $p_v$ increases to the saturation pressure $p_s$ and it is maximum partial pressure of water vapour at temperature $t_d$. The air containing moisture in such a state ( point $C$) is called *saturated air*.

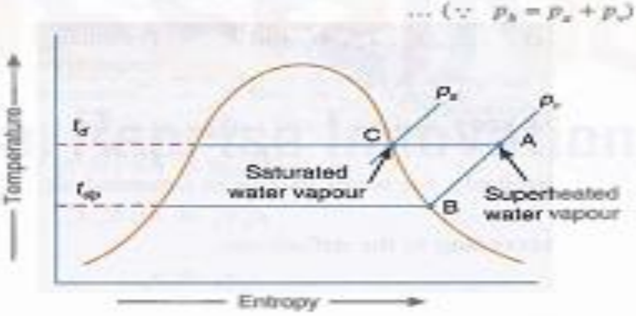


Fig. 16.3. $T$-$s$ diagram.

For saturated air (*i.e.* when the air is holding maximum amount of water vapour), the humidity ratio or maximum specific humidity,

$$W_s = W_{max} = 0.622 \times \frac{p_s}{p_b - p_s}$$

where $p_s$ = Partial pressure of air corresponding to saturation temperature (*i.e.* dry bulb temperature $t_d$).

**2. *Degree of saturation or percentage humidity*.** We have already discussed that the degree of saturation is the ratio of actual mass of water vapour in a unit mass of dry air to the mass of water vapour in the same mass of dry air when it is saturated at the same temperature (dry bulb temperature). In other words, it may be defined as the ratio of actual specific humidity to the specific humidity of saturated air at the same dry bulb temperature. It is, usually, denoted by $\mu$. Mathematically, degree of saturation,

$$\mu = \frac{W}{W_s} = \frac{\dfrac{0.622\, p_v}{p_b - p_v}}{\dfrac{0.622\, p_s}{p_b - p_s}} = \frac{p_v}{p_s}\left(\frac{p_b - p_s}{p_b - p_v}\right) = \frac{p_v}{p_s}\left[\frac{1 - \dfrac{p_s}{p_b}}{1 - \dfrac{p_v}{p_b}}\right]$$

**Notes :** **(*a*)** The partial pressure of saturated air ($p_s$) is obtained from the steam tables corresponding to dry bulb temperature $t_d$.

**(*b*)** If the relative humidity, $\phi = p_v / p_s$ is equal to zero, then the humidity ratio, $W = 0$, *i.e.* for dry air, $\mu = 0$.

**(*c*)** If the relative humidity, $\phi = p_v / p_s$ is equal to 1, then $W = W_s$ and $\mu = 1$. Thus $\mu$ varies between 0 and 1.

**3. *Relative humidity.*** We have already discussed that the relative humidity is the ratio of actual mass of water vapour ($m_v$) in a given volume of moist air to the mass of water vapour ($m_s$) in the same volume of saturated air at the same temperature and pressure. It is usually denoted by $\phi$. Mathematically, relative humidity,

$$\phi = \frac{m_v}{m_s}$$

Let $p_v$, $v_v$, $T_v$, $m_v$ and $R_v$ = Pressure, volume, temperature, mass and gas constant respectively for water vapour in actual conditions, and

$p_s$, $v_s$, $T_s$, $m_s$ and $R_s$ = Corresponding values for water vapour in saturated air.

We know that for water vapour in actual conditions,

$$p_v v_v = m_v R_v T_v \qquad \text{... (i)}$$

Similarly, for water vapour in saturated air,

$$p_s v_s = m_s R_s T_s \qquad \text{... (ii)}$$

According to the definitions,

$$v_v = v_s$$

and

$$T_v = T_s$$

Also

$$R_v = R_s = 0.461 \text{ kJ/kg K}$$

$\therefore$ From equations (i) and (ii), relative humidity,

$$\phi = \frac{m_v}{m_s} = \frac{p_v}{p_s}$$

Thus, the relative humidity may also be defined as the ratio of actual partial pressure of water vapour in moist air at a given temperature (dry bulb temperature) to the saturation pressure of water vapour (or partial pressure of water vapour in saturated air) at the same temperature.

The relative humidity may also be obtained as discussed below :

We know that degree of saturation,

$$\mu = \frac{p_v}{p_s}\left[\frac{1-\frac{p_s}{p_b}}{1-\frac{p_v}{p_b}}\right] = \phi\left[\frac{1-\frac{p_s}{p_b}}{1-\phi\times\frac{p_s}{p_b}}\right] \qquad \text{...} \left(\because \phi = \frac{p_v}{p_s}\right)$$

$\therefore$

$$\phi = \frac{\mu}{1-(1-\mu)\frac{p_s}{p_b}}$$

**Note :** For saturated air, the relative humidity is 100%.

**4. *Pressure of water vapour.*** According to Carrier's equation, the partial pressure of water vapour,

$$p_v = p_w - \frac{(p_b - p_w)(t_d - t_w)}{1544 - 1.44\, t_w}$$

where $p_w$ = Saturation pressure corresponding to wet bulb temperature (from steam tables),

$p_b$ = Barometric pressure,

$t_d$ = Dry bulb temperature, and

$t_w$ = Wet bulb temperature.

**5. *Vapour density or absolute humidity.*** We have already discussed that the vapour density or absolute humidity is the mass of water vapour present in 1 $m^3$ of dry air.

Let $v_v$ = Volume of water vapour in $m^3$/kg of dry air at its partial pressure,

$v_a$ = Volume of dry air in $m^3$/kg of dry air at its partial pressure,

$\rho_v$ = Density of water vapour in kg/$m^3$ corresponding to its partial pressure and dry bulb temperature $t_d$, and

$\rho_a$ = Density of dry air in kg/$m^3$ of dry air.

We know that mass of water vapour,

$$m_v = v_v \rho_v \qquad \ldots (i)$$

and mass of dry air,

$$m_a = v_a \rho_a \qquad \ldots (ii)$$

Dividing equation (*i*) by equation (*ii*),

$$\frac{m_v}{m_a} = \frac{v_v \rho_v}{v_a \rho_a}$$

Since $v_a = v_v$, therefore humidity ratio,

$$W = \frac{m_v}{m_a} = \frac{\rho_v}{\rho_a} \quad \text{or} \quad \rho_v = W \rho_a \qquad \ldots (iii)$$

We know that $p_a v_a = m_a R_a T_d$

Since $v_a = \frac{1}{\rho_a}$ and $m_a = 1$ kg, therefore substituting these values in the above expression, we get

$$p_a \times \frac{1}{\rho_a} = R_a T_d \quad \text{or} \quad \rho_a = \frac{p_a}{R_a T_d}$$

Substituting the value of $\rho_a$ in equation (*iii*), we have

$$\rho_v = \frac{W p_a}{R_a T_d} = \frac{W(p_b - p_v)}{R_a T_d} \qquad \ldots (\because\ p_b = p_a + p_v)$$

where $p_a$ = Pressure of air in kN/$m^2$,

$R_a$ = Gas constant for air = 0.287 kJ/ kg K, and

$T_d$ = Dry bulb temperature in K.

## 16.5 Enthalpy (Total Heat) of Moist Air

The enthalpy of moist air is numerically equal to the enthalpy of dry air plus the enthalpy of water vapour associated with dry air. Let us consider one kg of dry air. We know that enthalpy of 1 kg of dry air,

$$h_a = c_{pa}\, t_d \qquad \ldots (i)$$

where $c_{pa}$ = Specific heat of dry air which is normally taken as 1.005 kJ / kg K, and

$t_d$ = Dry bulb temperature.

Enthalpy of water vapour associated with 1 kg of dry air,

$$h_v = W\, h_s \qquad \ldots (ii)$$

where $W$ = Mass of water vapour in 1 kg of dry air (*i.e.* specific humidity), and

$h_s$ = Enthalpy of water vapour per kg of dry air at dew point temperature ($t_{dp}$).

If the moist air is superheated, then the enthalpy of water vapour

$$= W\, c_{ps}\, (t_d - t_{dp}) \qquad \ldots (iii)$$

where $c_{ps}$ = Specific heat of superheated water vapour which is normally taken as 1.9 kJ/kg K, and

$t_d - t_{dp}$ = Degree of superheat of the water vapour.

$\therefore$ Total enthalpy of superheated water vapour,

$$\begin{aligned}
h &= c_{pa}\, t_d + W\, h_s + W\, c_{ps}\, (t_d - t_{dp}) \\
&= c_{pa}\, t_d + W\, [h_{fdp} + h_{fgdp} + c_{ps}\, (t_d - t_{dp})] && \ldots (\because\ h_s = h_{fdp} + h_{fgdp}) \\
&= c_{pa}\, t_d + W\, [4.2\, t_{dp} + h_{fgdp} + c_{ps}\, (t_d - t_{dp})] && \ldots (\because\ h_{fdp} + 4.2\, t_{dp}) \\
&= c_{pa}\, t_d + 4.2\, W\, t_{dp} + W\, h_{fgdp} + W\, c_{ps}\, t_d - W\, c_{ps}\, t_{dp} \\
&= (c_{pa} + W\, c_{ps})\, t_d + W\, [h_{fgdp} + t_{dp}\, (4.2 - c_{ps})] \\
&= (c_{pa} + W\, c_{ps})\, t_d + W\, [h_{fgdp} + t_{dp}\, (4.2 - 1.9)] \\
&= (c_{pa} + W\, c_{ps})\, t_d + W\, [h_{fgdp} + 2.3\, t_{dp}]
\end{aligned}$$

The term $(c_{pa} + W\, c_{ps})$ is called *humid specific heat* ($c_{pm}$). It is the specific heat or heat capacity of moist air, *i.e.* $(1 + W)$ kg / kg of dry air. At low temperature of air conditioning range, the value of $W$ is very small. The general value of humid specific heat in air conditioning range is taken as 1.022 kJ/kg K.

$$\therefore \qquad h = 1.022\, t_d + W\, (h_{fgdp} + 2.3\, t_{dp}) \text{ kJ/kg}$$

where $h_{fgdp}$ = Latent heat of vaporisation of water corresponding to dew point temperature (from steam tables).

An approximate result may be obtained by the following relation:

$$h = 1.005\, t_d + W\, [2500 + 1.9\, t_d] \text{ kJ/kg}$$

**Example 16.1.** *The readings from a sling psychrometer are as follows :*

*Dry bulb temperature = 30° C ; Wet bulb temperature = 20° C ; Barometer reading = 740 mm of Hg.*

*Using steam tables, determine : 1. Dew point temperature ; 2. Relative humidity ; 3. Specific humidity ; 4. Degree of saturation ; 5. Vapour density ; and 6. Enthalpy of mixture per kg of dry air.*

**Solution.** Given : $t_d = 30$°C ; $t_w = 20$°C ; $p_b = 740$ mm of Hg

*1. Dew point temperature*

First of all, let us find the partial pressure of water vapour ($p_v$).

From steam tables, we find that the saturation pressure corresponding to wet bulb temperature of 20° C is

$$p_w = 0.023\ 37 \text{ bar}$$

We know that barometric pressure,

$$p_b = 740 \text{ mm of Hg} \quad \text{.... (Given)}$$

$$= 740 \times 133.3 = 98\ 642 \text{ N/m}^2 \quad \text{... } (\because 1 \text{ mm of Hg} = 133.3 \text{ N/m}^2)$$

$$= 0.986\ 42 \text{ bar} \quad \text{.... } (\because 1 \text{ bar} = 10^5 \text{ N/m}^2)$$

∴ Partial pressure of water vapour,

$$p_v = p_w - \frac{(p_b - p_w)(t_d - t_w)}{1544 - 1.44\, t_w}$$

$$= 0.023\ 37 - \frac{(0.986\ 42 - 0.02337)(30 - 20)}{1544 - 1.44 \times 20}$$

$$= 0.023\ 37 - 0.006\ 36 = 0.017\ 01 \text{ bar}$$

Since the dew point temperature is the saturation temperature corresponding to the partial pressure of water vapour ($p_v$), therefore from steam tables, we find that corresponding to a pressure of 0.017 01 bar, the dew point temperature is

$$t_{dp} = 15° \text{ C} \quad \textbf{Ans.}$$

*2. Relative humidity*

From steam tables, we find that the saturation pressure of vapour corresponding to dry bulb temperature of 30°C is

$$p_s = 0.042\ 42 \text{ bar}$$

We know that relative humidity,

$$\phi = \frac{p_v}{p_s} = \frac{0.01701}{0.04242} = 0.40 \text{ or } 40\% \quad \textbf{Ans.}$$

*3. Specific humidity*

We know that specific humidity,

$$W = \frac{0.622\, p_v}{p_b - p_v} = \frac{0.622 \times 0.01701}{0.986\ 42 - 0.01701}$$

$$= \frac{0.01058}{0.96941} = 0.010\ 914 \text{ kg/kg of dry air}$$

$$= 10.914 \text{ g/kg of dry air} \quad \textbf{Ans.}$$

*4. Degree of saturation*

We know that specific humidity of saturated air,

$$W_s = \frac{0.622\, p_s}{p_b - p_s} = \frac{0.622 \times 0.04242}{0.98642 - 0.04242}$$

$$= \frac{0.02638}{0.944} = 0.027\ 945 \text{ kg/kg of dry air}$$

We know that degree of saturation,

$$\mu = \frac{W}{W_s} = \frac{0.010\ 914}{0.027\ 945} = 0.391 \text{ or } 39.1\% \quad \textbf{Ans.}$$

**Note :** The degree of saturation (μ) may also be calculated from the following relation :

$$\mu = \frac{p_v}{p_s}\left(\frac{p_b - p_s}{p_b - p_v}\right)$$

$$= \frac{0.01701}{0.04242}\left[\frac{0.98642 - 0.04242}{0.98642 - 0.01701}\right]$$

$$= 0.391 \text{ or } 39.1\% \quad \textbf{Ans.}$$

5. *Vapour density*

We know that vapour density,

$$\rho_v = \frac{W(p_b - p_v)}{R_a T_d} = \frac{0.010\,914\,(0.98642 - 0.01701)\,10^5}{287\,(273+30)}$$

$$= 0.012\,16 \text{ kg/m}^3 \text{ of dry air} \quad \textbf{Ans.}$$

6. *Enthalpy of mixture per kg of dry air*

From steam tables, we find that the latent heat of vaporisation of water at dew point temperature of 15°C is

$$h_{fgdp} = 2466.1 \text{ kJ/kg}$$

∴ Enthalpy of mixture per kg of dry air,

$$h = 1.022\, t_d + W\,[h_{fgdp} + 2.3\, t_{dp}]$$

$$= 1.022 \times 30 + 0.010\,914\,[2466.1 + 2.3 \times 15]$$

$$= 30.66 + 27.29 = 57.95 \text{ kJ/kg of dry air} \quad \textbf{Ans.}$$

**Example 16.2.** *On a particular day, the atmospheric air was found to have a dry bulb temperature of 30° C and a wet bulb temperature of 18° C. The barometric pressure was observed to be 756 mm of Hg. Using the tables of psychrometric properties of air, determine the relative humidity, the specific humidity, the dew point temperature, the enthalpy of air per kg of dry air and the volume of mixture per kg of dry air.*

**Solution.** Given : $t_d = 30°C$ ; $t_w = 18°C$ ; $p_b = 756$ mm of Hg

*Relative humidity*

First of all, let us find the partial pressure of water vapour ($p_v$). From steam tables, we find that the saturation pressure corresponding to wet bulb temperature of 18°C is,

$$p_w = 0.020\,62 \text{ bar} = 0.020\,62 \times 10^5 = 2062 \text{ N/m}^2$$

$$= \frac{2062}{133.3} = 15.47 \text{ mm of Hg} \qquad \ldots (\because 1 \text{ mm of Hg} = 133.3 \text{ N/m}^2)$$

We know that

$$p_v = p_w - \frac{(p_b - p_w)(t_d - t_w)}{1544 - 1.44\, t_w}$$

$$= 15.47 - \frac{(756 - 15.47)(30 - 18)}{1544 - 1.44 \times 18} \text{ mm of Hg}$$

$$= 15.47 - 5.85 = 9.62 \text{ mm of Hg}$$

From steam tables, we find that the saturation pressure of vapour corresponding to dry bulb temperature of 30°C is

$$p_s = 0.042\,42 \text{ bar} = 0.042\,42 \times 10^5 = 4242 \text{ N/m}^2$$

$$= \frac{4242}{133.3} = 31.8 \text{ mm of Hg}$$

We know that the relative humidity,

$$\phi = \frac{p_v}{p_s} = \frac{9.62}{31.8} = 0.3022 \text{ or } 30.22\% \quad \textbf{Ans.}$$

*Specific humidity*

We know that specific humidity,

$$W = \frac{0.622\, p_v}{p_b - p_v} = \frac{0.622 \times 9.62}{756 - 9.62} = 0.008 \text{ kg/kg of dry air} \quad \textbf{Ans.}$$

*Dew point temperature*

Since the dew point temperature is the saturation temperature corresponding to the partial pressure of water vapour ($p_v$), therefore from steam tables, we find that corresponding to 9.62 mm of Hg or $9.62 \times 133.3 = 1282.3 \text{ N/m}^2 = 0.012\ 823$ bar, the dew point temperature is,

$$t_{dp} = 10.6°\text{ C} \quad \textbf{Ans.}$$

*Enthalpy of air per kg of dry air*

From steam tables, we also find that latent heat of vaporisation of water at dew point temperature of 10.6°C,

$$h_{fgdp} = 2476.5 \text{ kJ/kg}$$

We know that enthalpy of air per kg of dry air,

$$h = 1.022\, t_d + W\,(h_{fgdp} + 2.3\, t_{dp})$$

$$= 1.022 \times 30 + 0.008\,(2476.5 + 2.3 \times 10.6)$$

$$= 30.66 + 20 = 50.66 \text{ kJ/kg of dry air} \quad \textbf{Ans.}$$

*Volume of the mixture per kg of dry air*

From psychrometric tables, we find that specific volume of the dry air at 760 mm of Hg and 30°C dry bulb temperature is 0.8585 m³/kg of dry air. We know that one kg of dry air at a partial pressure of (756 – 9.62) mm of Hg occupies the same volume as $W = 0.008$ kg of vapour at its partial pressure of 9.62 mm of Hg. Moreover, the mixture occupies the same volume but at a total pressure of 756 mm of Hg.

∴ Volume of the mixture ($v$) at a dry bulb temperature of 30°C and a pressure of 9.62 mm of Hg

= Volume of 1 kg of dry air ($v_a$) at a pressure of ( 756 – 9.62 ) or 746.38 mm of Hg

$$= 0.8585 \times \frac{760}{746.38} = 0.8741 \text{ kg/kg of dry air} \quad \textbf{Ans.}$$

**Note :** The volume of mixture per kg of dry air may be calculated as discussed below :

We know that $\quad v = v_a = \dfrac{R_a\, T_d}{p_a}$

where

$R_a$ = Gas constant for air = 287 J/kg K

$T_d$ = Dry bulb temperature in K

$= 30 + 273 = 303$ K, and

$p_a$ = Pressure of air in N/m²

$= p_b - p_v = 756 - 9.62 = 746.38$ mm of Hg

$= 746.38 \times 133.3 = 994\ 92 \text{ N/m}^2$

Substituting the values in the above equation,

$$v = \frac{287 \times 303}{994\,92} = 0.8741 \text{ m}^3\text{/kg of dry air} \quad \textbf{Ans.}$$

## 16.6 Thermodynamic Wet Bulb Temperature or Adiabatic Saturation Temperature

The thermodynamic wet bulb temperature or adiabatic saturation temperature is the temperature at which the air can be brought to saturation state, adiabatically, by the evaporation of water into the flowing air.

The equipment used for the adiabatic saturation of air, in its simplest form, consists of an insulated chamber containing adequate quantity of water. There is also an arrangement for extra water (known as make-up water) to flow into the chamber from its top, as shown in Fig. 16.4.

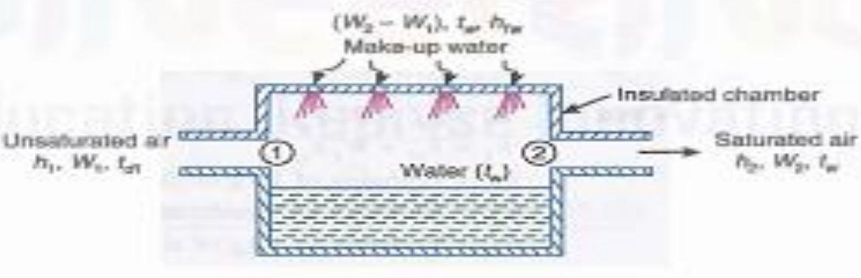



**Fig. 16.4.** Adiabatic saturation of air.

Let the unsaturated air enters the chamber at section 1. As the air passes through the chamber over a long sheet of water, the water evaporates which is carried with the flowing stream of air, and the specific humidity of the air increases. The make-up water is added to the chamber at this temperature to make the water level constant. Both the air and water are cooled as the evaporation takes place. This process continues until the energy transferred from the air to the water is equal to the energy required to vaporise the water. When steady conditions are reached, the air flowing at section 2 is saturated with water vapour. The temperature of the saturated air at section 2 is known as *thermodynamic wet bulb temperature* or *adiabatic saturation temperature*.

The adiabatic saturation process can be represented on *T-s* diagram as shown by the curve 1-2 in Fig. 16.5.

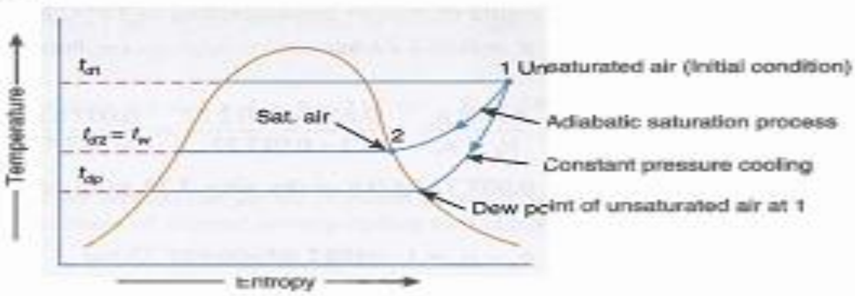



**Fig. 16.5.** *T-s* diagram for adiabatic saturation process.

During the adiabatic saturation process, the partial pressure of vapour increases, although the total pressure of the air-vapour mixture remains constant. The unsaturated air initially at dry bulb temperature $t_{d1}$ is cooled adiabatically to dry bulb temperature $t_{d2}$ which is equal to the adiabatic saturation temperature $t_w$. It may be noted that the adiabatic saturation temperature is taken equal to the wet bulb temperature for all practical purposes.

Let $h_1$ = Enthalpy of unsaturated air at section 1,

$W_1$ = Specific humidity of air at section 1,

$h_2$, $W_2$ = Corresponding values of saturated air at section 2, and

$h_{fw}$ = Sensible heat of water at adiabatic saturation temperature.

Balancing the enthalpies of air at inlet and outlet (*i.e.* at sections 1 and 2),

$$h_1 + (W_2 - W_1)\, h_{fw} = h_2 \qquad \ldots (i)$$

or

$$h_1 - W_1\, h_{fw} = h_2 - W_2\, h_{fw} \qquad \ldots (ii)$$

The term $(h_2 - W_2\, h_{fw})$ is known as *sigma heat* and remains constant during the adiabatic process.

We know that $h_1 = h_{a1} + W_1\, h_{s1}$

and $h_2 = h_{a2} + W_2\, h_{s2}$

where $h_{a1}$ = Enthalpy of 1 kg of dry air at dry bulb temperature $t_{d1}$,

*$h_{s1}$ = Enthalpy of superheated vapour at $t_{d1}$ per kg of vapour,

$h_{a2}$ = Enthalpy of 1 kg of air at wet bulb temperature $t_w$, and

$h_{s2}$ = Enthalpy of saturated vapour at wet bulb temperature $t_w$ per kg of vapour.

Now the equation (*ii*) may be written as :

$$(h_{a1} + W_1\, h_{s1}) - W_1\, h_{fw} = (h_{a2} + W_2\, h_{s2}) - W_2\, h_{fw}$$

$$W_1\,(h_{s1} - h_{fw}) = W_2\,(h_{s2} - h_{fw}) + h_{a2} - h_{a1}$$

$$\therefore \quad W_1 = \frac{W_2\,(h_{s2} - h_{fw}) + h_{a2} - h_{a1}}{h_{s1} - h_{fw}}$$

## 16.7 Psychrometric Chart

It is a graphical representation of the various thermodynamic properties of moist air. The psychrometric chart is very useful for finding out the properties of air (which are required in the field of air conditioning) and eliminate lot of calculations. There is a slight variation in the charts prepared by different air-conditioning manufactures but basically they are all alike. The psychrometric chart is normally drawn for standard atmospheric pressure of 760 mm of Hg (or 1.01325 bar).

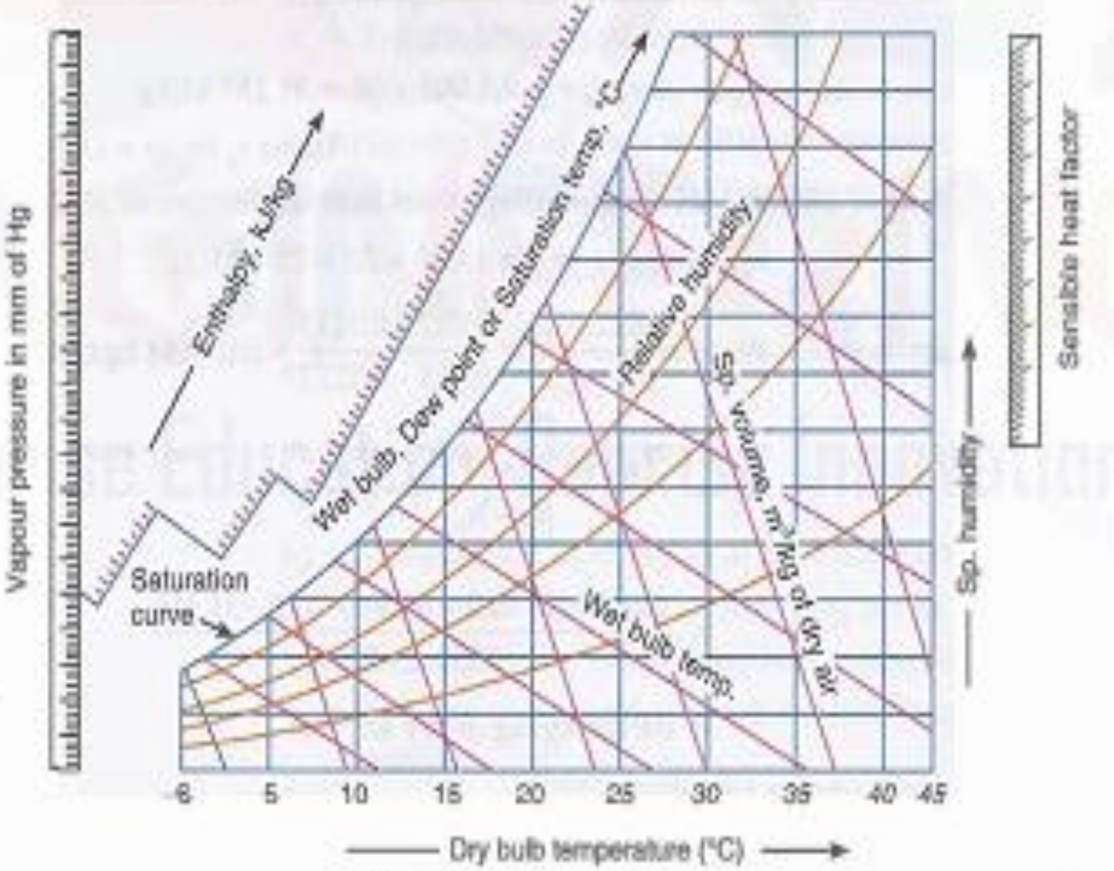



Fig. 16.6. Psychrometric chart.

In a psychrometric chart, dry bulb temperature is taken as abscissa and specific humidity *i.e.* moisture contents as ordinate, as shown in Fig. 16.6. Now the saturation curve is drawn by plotting the various saturation points at corresponding dry bulb temperatures. The saturation curve represents 100% relative humidity at various dry bulb temperatures. It also represents the wet bulb and dew point temperatures.

Though the psychrometric chart has a number of details, yet the following lines are important from the subject point of view :

**1. *Dry bulb temperature lines.*** The dry bulb temperature lines are vertical *i.e.* parallel to the ordinate and uniformly spaced as shown in Fig. 16.7. Generally the temperature range of these lines on psychrometric chart is from – 6° C to 45° C. The dry bulb temperature lines are drawn with difference of every 5°C and up to the saturation curve as shown in the figure. The values of dry bulb temperatures are also shown on the saturation curve.

**2. *Specific humidity or moisture content lines.*** The specific humidity (moisture content) lines are horizontal *i.e.* parallel to the abscissa and are also uniformly spaced as shown in Fig. 16.8. Generally, moisture content range of these lines on psychrometric chart is from 0 to 30 g / kg of dry air (or from 0 to 0.030 kg / kg of dry air). The moisture content lines are drawn with a difference of every 1 g (or 0.001 kg) and up to the saturation curve as shown in the figure.

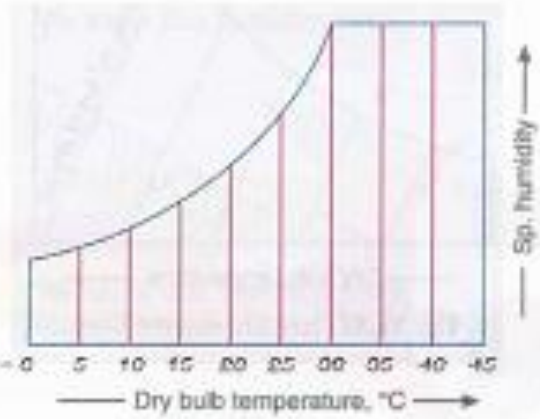



Fig. 16.7. Dry bulb temperature lines.

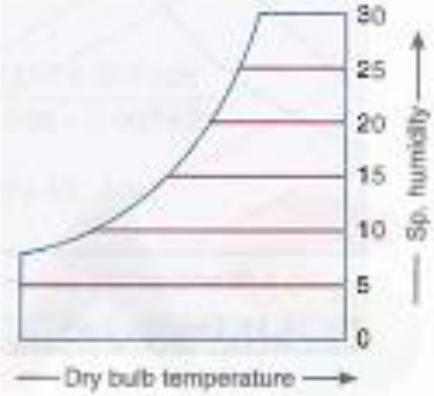



Fig. 16.8. Specific humidity lines.

**3. *Dew point temperature lines.*** The dew point temperature lines are horizontal *i.e.* parallel to the abscissa and non-uniformly spaced as shown in Fig. 16.9. At any point on the saturation curve, the dry bulb and dew point temperatures are equal.

The values of dew point temperatures are generally given along the saturation curve of the chart as shown in the figure.

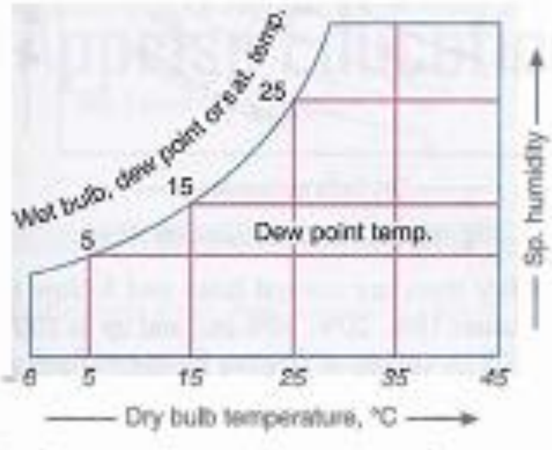



Fig. 16.9. Dew point temperature lines.

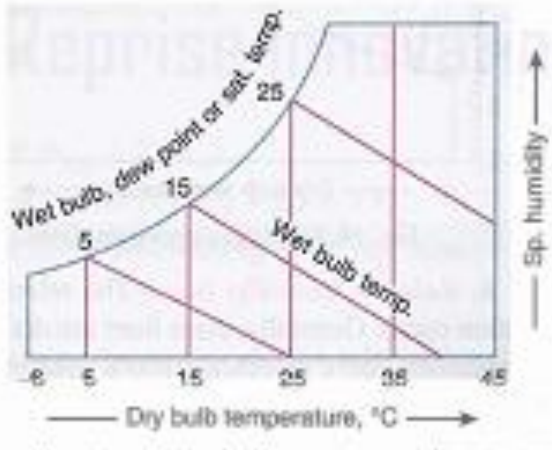



Fig. 16.10. Wet bulb temperature lines.

**4. *Wet bulb temperature lines.*** The wet bulb temperature lines are inclined straight lines and non-uniformly spaced as shown in Fig. 16.10. At any point on the saturation curve, the dry bulb and wet bulb temperatures are equal.

The values of wet bulb temperatures are generally given along the saturation curve of the chart as shown in the figure.

**5.** ***Enthalpy (total heat) lines.*** The enthalpy (or total heat) lines are inclined straight lines and uniformly spaced as shown in Fig. 16.11. These lines are parallel to the wet bulb temperature lines, and are drawn up to the saturation curve. Some of these lines coincide with the wet bulb temperature lines also.

The values of total enthalpy are given on a scale above the saturation curve as shown in the figure.

**6.** ***Specific volume lines.*** The specific volume lines are obliquely inclined straight lines and uniformly spaced as shown in Fig. 16.12. These lines are drawn up to the saturation curve.

The values of volume lines are generally given at the base of the chart.

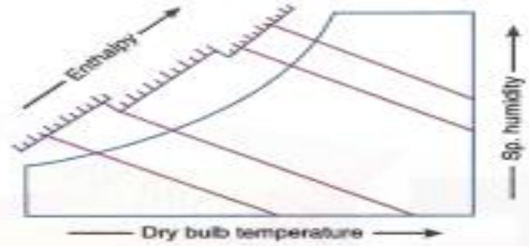



**Fig. 16.11.** Enthalpy lines.

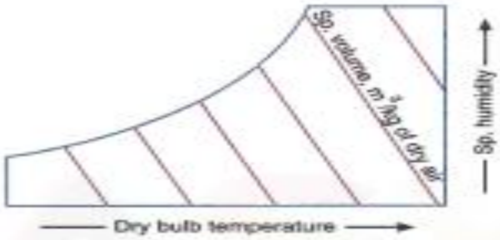



**Fig. 16.12.** Specific volume lines.

**7.** ***Vapour pressure lines.*** The vapour pressure lines are horizontal and uniformly spaced. Generally, the vapour pressure lines are not drawn in the main chart. But a scale showing vapour pressure in mm of Hg is given on the extreme left side of the chart as shown in Fig. 16.13.

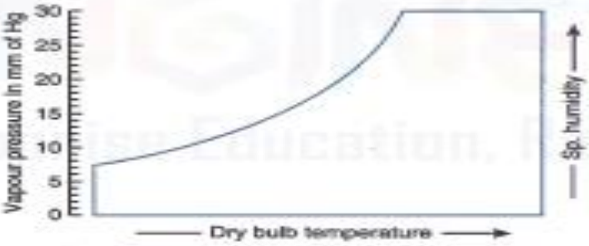



**Fig. 16.13.** Vapour pressure lines.

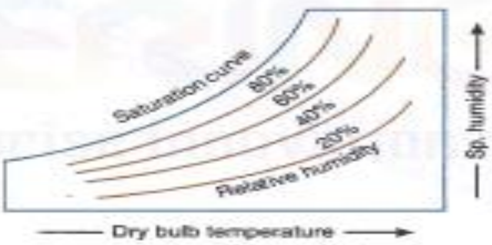



**Fig. 16.14.** Relative humidity lines.

**8.** ***Relative humidity lines.*** The relative humidity lines are curved lines and follow the saturation curve. Generally, these lines are drawn with values 10%, 20%, 30% etc. and up to 100%. The saturation curve represents 100% relative humidity. The values of relative humidity lines are generally given along the lines themselves as shown in Fig. 16.14.

## 16.8 Psychrometric Processes

The various psychrometric processes involved in air conditioning to vary the psychrometric properties of air according to the requirement are as follows :

1. Sensible heating, 2. Sensible cooling, 3. Humidification and dehumidification, 4. Cooling and adiabatic humidification, 5. Cooling and humidification by water injection, 6. Heating and humidification, 7. Humidification by steam injection, 8. Adiabatic chemical dehumidification, 9. Adiabatic mixing of air streams.

We shall now discuss these psychrometric processes, in detail, in the following pages.

## 16.9 Sensible Heating

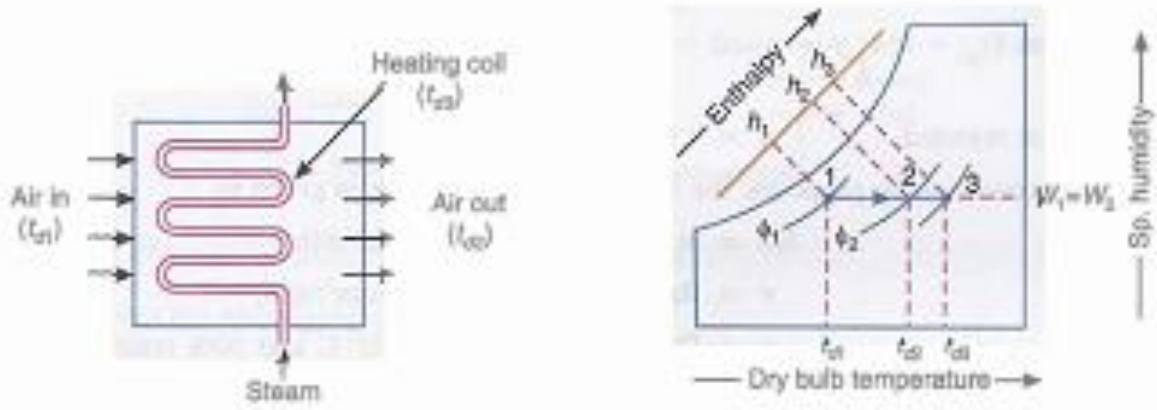
The heating of air, without any change in its specific humidity, is known as *sensible heating*. Let air at temperature $t_{d1}$ passes over a heating coil of temperature $t_{d3}$, as shown in Fig. 16.16 (*a*). It may be noted that the temperature of air leaving the heating coil ($t_{d2}$) will be less than $t_{d3}$. The process of sensible heating, on the psychrometric chart, is shown by a horizontal line 1-2 extending from left to right as shown in Fig. 16.16 (*b*). The point 3 represents the surface temperature of the heating coil.

The heat absorbed by the air during sensible heating may be obtained from the psychrometric chart by the enthalpy difference ($h_2 - h_1$) as shown in Fig. 16.16 (*b*). It may be noted that the specific humidity during the sensible heating remains constant (*i.e.* $W_1 = W_2$). The dry bulb temperature increases from $t_{d1}$ to $t_{d2}$ and relative humidity reduces from $\phi_1$ to $\phi_2$ as shown in Fig. 16.16 (*b*). The amount of heat added during sensible heating may also be obtained from the relation :

Heat added,
$$q = h_2 - h_1$$
$$= c_{pa}\,(t_{d2} - t_{d1}) + W\,c_{ps}\,(t_{d2} - t_{d1})$$
$$= (c_{pa} + W\,c_{ps})\,(t_{d2} - t_{d1}) = c_{pm}\,(t_{d2} - t_{d1})$$

The term $(c_{pa} + W\, c_{ps})$ is called *humid specific heat* $(c_{pm})$ and its value is taken as 1.022 kJ /kg K.

$\therefore$ Heat added, $\quad q = 1.022\,(t_{d2} - t_{d1})$ kJ/kg




(*a*) Psychrometric process. (*b*) Psychrometric chart.

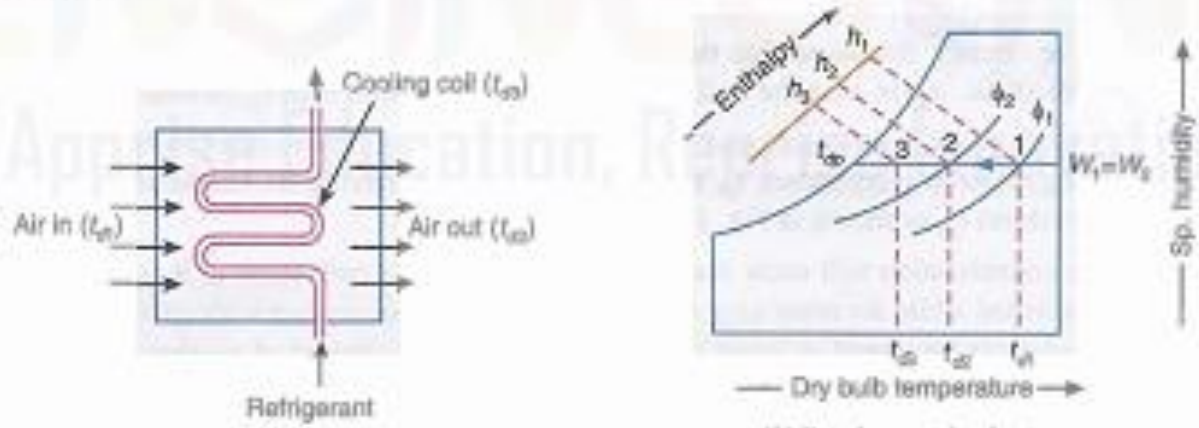
**Fig. 16.16.** Sensible heating.

**Notes : 1.** For sensible heating, steam or hot water is passed through the heating coil. The heating coil may be electric resistance coil.

**2.** The sensible heating of moist air can be done to any desired temperature.

## 16.10 Sensible Cooling

The cooling of air, without any change in its specific humidity, is known as *sensible cooling*. Let air at temperature $t_{d1}$ passes over a cooling coil of temperature $t_{d3}$ as shown in Fig. 16.17 (*a*). It may be noted that the temperature of air leaving the cooling coil $(t_{d2})$ will be more than $t_{d3}$. The process of sensible cooling, on the psychrometric chart, is shown by a horizontal line 1-2 extending from right to left as shown in Fig. 16.17 (*b*). The point 3 represents the surface temperature of the cooling coil.




(*a*) Psychrometric process. (*b*) Psychrometric chart.

**Fig. 16.17.** Sensible cooling.

The heat rejected by air during sensible cooling may be obtained from the psychrometric chart by the enthalpy difference $(h_1 - h_2)$ as shown in Fig. 16.17 (*b*).

It may be noted that the specific humidity during the sensible cooling remains constant (*i.e.* $W_1 = W_2$). The dry bulb temperature reduces from $t_{d1}$ to $t_{d2}$ and relative humidity increases from $\phi_1$ to $\phi_2$, as shown in Fig. 16.17 (*b*). The amount of heat rejected during sensible cooling may also be obtained from the relation :

Heat rejected, $q = h_1 - h_2$

$$= c_{pa}(t_{d1} - t_{d2}) + W\,c_{ps}(t_{d1} - t_{d2})$$

$$= (c_{pa} + W\,c_{ps})(t_{d1} - t_{d2}) = c_{pm}(t_{d1} - t_{d2})$$

The term $(c_{pa} + W\,c_{ps})$ is called *humid specific heat* ($c_{pm}$) and its value is taken as 1.022 kJ /kg K.

$\therefore$ Heat rejected, $q = 1.022\,(t_{d1} - t_{d2})$ kJ/kg

For air conditioning purposes, the sensible heat per minute is given as

$$SH = m_a\,c_{pm}\,\Delta t = v\,\rho\,c_{pm}\,\Delta t \text{ kJ/min} \qquad \ldots (\because\ m = v\,\rho)$$

where $v$ = Rate of dry air flowing in m$^3$/min,

$\rho$ = Density of moist air at 20° C and 50% relative humidity = 1.2 kg / m$^3$ of dry air,

$c_{pm}$ = Humid specific heat = 1.022 kJ /kg K, and

$\Delta t$ = $t_{d1} - t_{d2}$ = Difference of dry bulb temperatures between the entering and leaving conditions of air in ° C.

Substituting the values of $\rho$ and $c_{pm}$ in the above expression, we get

$$SH = v \times 1.2 \times 1.022 \times \Delta t = 1.2264\, v \times \Delta t \text{ kJ/min}$$

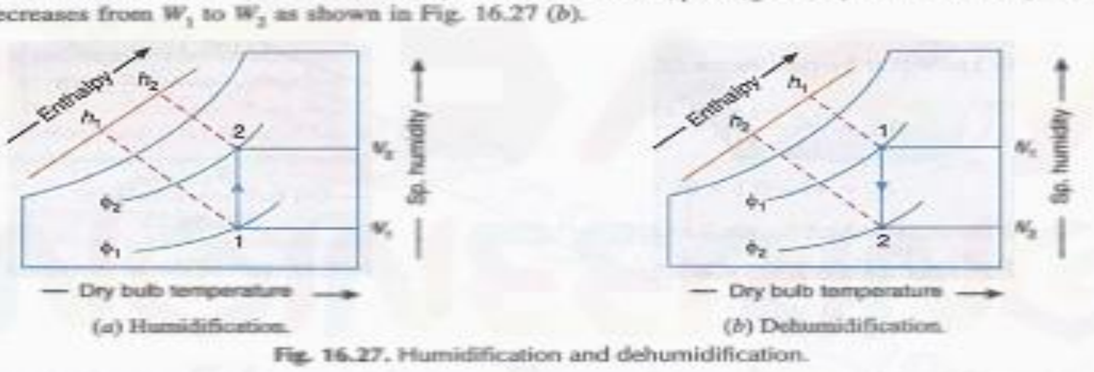
$$= \frac{1.2264\, v \times \Delta t}{60} = 0.020\,44\, v \times \Delta t \text{ kJ/s or kW} \qquad \ldots (\because\ 1 \text{ kJ/s} = 1 \text{ kW})$$

## 16.13 Humidification and Dehumidification

The addition of moisture to the air, without change in its dry bulb temperature, is known as *humidification*. Similarly, removal of moisture from the air, without change in its dry bulb temperature, is known as *dehumidification*. The heat added during humidification process and heat removed during dehumidification process is shown on the psychrometric chart in Fig. 16.27 (*a*) and (*b*) respectively.

Ultrasonic humidification system.

It may be noted that in humidification, the relative humidity increases from $\phi_1$ to $\phi_2$ and specific humidity also increases from $W_1$ to $W_2$ as shown in Fig. 16.27 (*a*). Similarly, in dehumidification, the relative humidity decreases from $\phi_1$ to $\phi_2$ and specific humidity also decreases from $W_1$ to $W_2$ as shown in Fig. 16.27 (*b*).



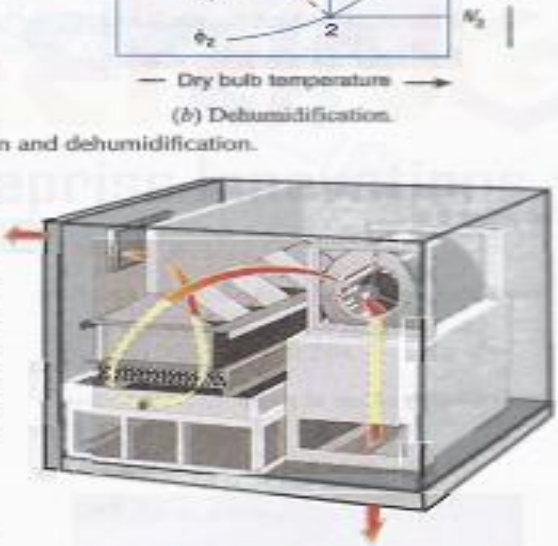

(*a*) Humidification. (*b*) Dehumidification.

**Fig. 16.27.** Humidification and dehumidification.

It may be noted that in humidification, change in enthalpy is shown by the intercept $(h_2 - h_1)$ on the psychrometric chart. Since the dry bulb temperature of air during the humidification remains constant, therefore its sensible heat also remains constant. It is thus obvious that the change in enthalpy per kg of dry air due to the increased moisture content equal to $(W_2 - W_1)$ kg per kg of dry air is considered to cause a latent heat transfer (*LH*). Mathematically,

$$LH = (h_2 - h_1) = h_{fg}\,(W_2 - W_1)$$

where $h_{fg}$ is the latent heat of vaporisation at dry bulb temperature ($t_{d1}$).



Multiple small plate dehumidification system

## 16.15 Sensible Heat Factor

As a matter of fact, the heat added during a psychrometric process may be split up into sensible heat and latent heat. The ratio of the *sensible heat to the total heat is known as *sensible heat factor* (briefly written as *SHF*) or *sensible heat ratio* (briefly written as *SHR*). Mathematically,

$$SHF = \frac{\text{Sensible heat}}{\text{Total heat}} = \frac{SH}{SH + LH}$$

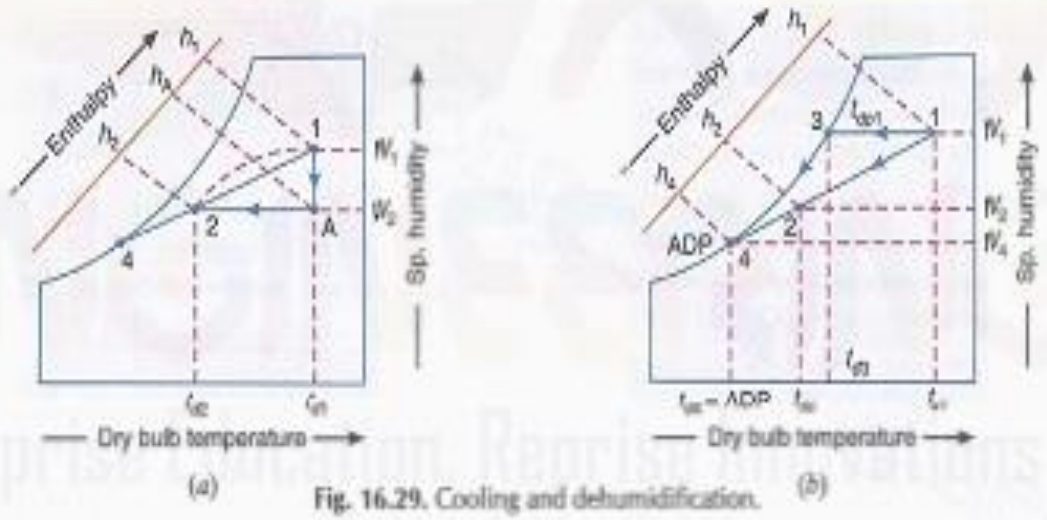
where $SH$ = Sensible heat, and

$LH$ = Latent heat.

The sensible heat factor scale is shown on the right hand side of the psychrometric chart.

## 16.16 Cooling and Dehumidification

This process is generally used in summer air conditioning to cool and dehumidify the air. The air is passed over a cooling coil or through a cold water spray. In this process, the dry bulb temperature as well as the specific humidity of air decreases. The final relative humidity of the air is generally higher than that of the entering air. The dehumidification of air is only possible when the effective surface temperature of the cooling coil (*i.e.* $t_{d4}$) is less than the dew point temperature of the air entering the coil (*i.e.* $t_{dp1}$). The effective surface temperature of the coil is known as *apparatus dew point* (briefly written as *ADP*). The cooling and dehumidification process is shown in Fig. 16.29.




(*a*) **Fig. 16.29.** Cooling and dehumidification. (*b*)

Let $t_{d1}$ = Dry bulb temperature of air entering the coil,

$t_{dp1}$ = Dew point temperature of the entering air = $t_{d3}$, and

$t_{d4}$ = Effective surface temperature or *ADP* of the coil.

Under ideal conditions, the dry bulb temperature of the air leaving the cooling coil (*i.e.* $t_{d4}$) should be equal to the surface temperature of the cooling coil (*i.e. ADP*), but it is never possible due to inefficiency of the cooling coil. Therefore, the resulting condition of air coming out of the coil is shown by a point 2 on the straight line joining the points 1 and 4. The by-pass factor in this case is given by

$$BPF = \frac{t_{d2} - t_{d4}}{t_{d1} - t_{d4}} = \frac{t_{d2} - ADP}{t_{d1} - ADP}$$

Also

$$BPF = \frac{W_2 - W_4}{W_1 - W_4} = \frac{h_2 - h_4}{h_1 - h_4}$$

Actually, the cooling and dehumidification process follows the path as shown by a dotted curve in Fig. 16.29 (*a*), but for the calculation of psychrometric properties, only end points are important. Thus the cooling and dehumidification process shown by a line 1-2 may be assumed to have followed a path 1-*A* (*i.e.* dehumidification) and *A*-2 (*i.e.* cooling) as shown in Fig. 16.29 (*a*). We see that the total heat removed from the air during the cooling and dehumidification process is

$$q = h_1 - h_2 = (h_1 - h_A) + (h_A - h_2) = LH + SH$$

where $LH = h_1 - h_A$ = Latent heat removed due to condensation of vapour of the reduced moisture content $(W_1 - W_2)$, and

$SH = h_A - h_2$ = Sensible heat removed.

We know that sensible heat factor,

$$SHF = \frac{\text{Sensible heat}}{\text{Total heat}} = \frac{SH}{LH + SH} = \frac{h_A - h_2}{h_1 - h_2}$$

**Note :** The line 1-4 (*i.e.* the line joining the point of entering air and the apparatus dew point) in Fig. 16.29 (*b*) is known as *sensible heat factor line*.

**Example 16.13.** *In a cooling application, moist air enters a refrigeration coil at the rate of 100 kg of dry air per minute at 35° C and 50% RH. The apparatus dew point of coil is 5° C and by-pass factor is 0.15. Determine the outlet state of moist air and cooling capacity of coil in TR.*

**Solution.** Given : $m_a = 100$ kg/min ; $t_{d1} = 35°$ C ; $\phi_1 = 50\%$ ; $ADP = 5°$ C ; $BPF = 0.15$

*Outlet state of moist air*

Let $t_{d2}$ and $\phi_2$ = Temperature and relative humidity of air leaving the cooling coil.

First of all, mark the initial condition of air, *i.e.* 35° C dry bulb temperature and 50% relative humidity on the psychrometric chart at point 1, as shown in Fig. 16.30. From the psychrometric chart, we find that the dew point temperature of the entering air at point 1,

$$t_{dp1} = 23° \text{ C}$$

Since the coil or apparatus dew point (*ADP*) is less than the dew point temperature of entering air, therefore it is a process of cooling and dehumidification.

We know that by-pass factor,

$$BPF = \frac{t_{d2} - t_{d4}}{t_{d1} - t_{d4}} = \frac{t_{d2} - ADP}{t_{d1} - ADP}$$

$$0.15 = \frac{t_{d2} - 5}{35 - 5} = \frac{t_{d2} - 5}{30}$$

$$\therefore \quad t_{d2} = 0.15 \times 30 + 5 = 9.5° \text{ C } \textbf{Ans.}$$

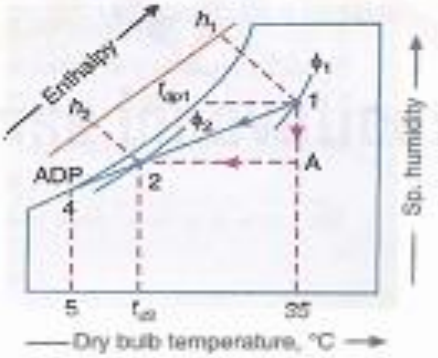



Fig. 16.30

From the psychrometric chart, we find that the relative humidity corresponding to a dry bulb temperature ($t_{d2}$) of 9.5° C on the line 1-4 is $\phi_2 = 99\%$. **Ans.**

dehumidification) and *A*-2 (*i.e.* cooling). Now from the psychrometric chart, we find that enthalpy of entering air at point 1,

$$h_1 = 81 \text{ kJ/kg of dry air}$$

and enthalpy of air at point 2,

$$h_2 = 28 \text{ kJ/kg of dry air}$$

We know that cooling capacity of the coil

$$= m_a (h_1 - h_2) = 100 (81 - 28) = 5300 \text{ kJ/min}$$

$$= 5300/210 = 25.24 \text{ TR } \textbf{Ans.} \qquad \ldots (\because 1 \text{ TR} = 210 \text{ kJ/min})$$

## 16.22 Adiabatic Mixing of Two Air Streams

When two quantities of air having different enthalpies and different specific humidities are mixed, the final condition of the air mixture depends upon the masses involved, and on the enthalpy and specific humidity of each of the constituent masses which enter the mixture.

Now consider two air streams 1 and 2 mixing adiabatically as shown in Fig. 16.52 (*a*).

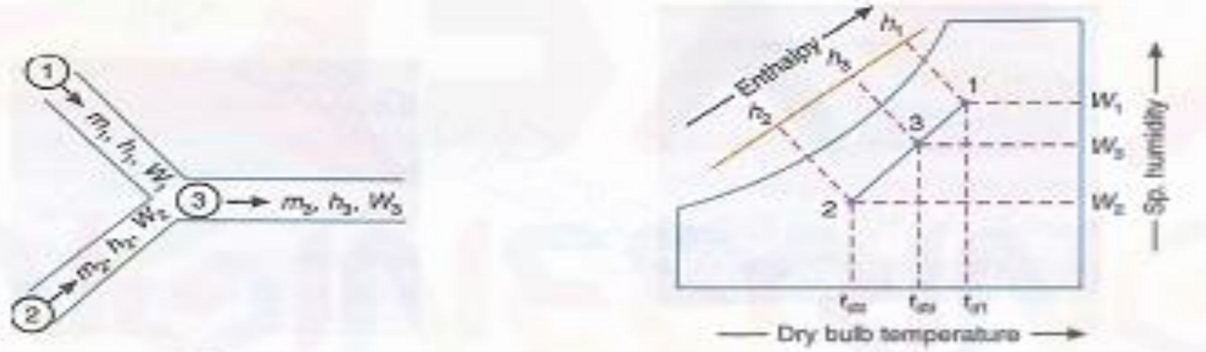
Let $m_1$ = Mass of air entering at 1,

$h_1$ = Enthalpy of air entering at 1,

$W_1$ = Specific humidity of air entering at 1,

$m_2, h_2, W_2$ = Corresponding values of air entering at 2, and

$m_3, h_3, W_3$ = Corresponding values of the mixture leaving at 3.




(*a*) (*b*)

**Fig. 16.52.** Adiabatic mixing of two air streams.

Assuming no loss of enthalpy and specific humidity during the air mixing process, we have for the mass balance,

$$m_1 + m_2 = m_3 \quad \ldots (i)$$

For the energy balance,

$$m_1 h_1 + m_2 h_2 = m_3 h_3 \quad \ldots (ii)$$

and for the mass balance of water vapour,

$$m_1 W_1 + m_2 W_2 = m_3 W_3 \quad \ldots (iii)$$

Substituting the value of $m_3$ from equation (*i*) in equation (*ii*),

$$m_1 h_1 + m_2 h_2 = (m_1 + m_2) h_3 = m_1 h_3 + m_2 h_3$$

or

$$m_1 h_1 - m_1 h_3 = m_2 h_3 - m_2 h_2$$

$$m_1 (h_1 - h_3) = m_2 (h_3 - h_2)$$

$$\therefore \quad \frac{m_1}{m_2} = \frac{h_3 - h_2}{h_1 - h_3} \quad \ldots (iv)$$

Similarly, substituting the value of $m_3$ from equation (*i*) in equation (*iii*), we have

$$\frac{m_1}{m_2} = \frac{W_3 - W_2}{W_1 - W_3} \quad \ldots (v)$$

Now from equations (*iv*) and (*v*),

$$\frac{m_1}{m_2} = \frac{h_3 - h_2}{h_1 - h_3} = \frac{W_3 - W_2}{W_1 - W_3} \quad \ldots (vi)$$

## 17.5 Effective Temperature

The degree of warmth or cold felt by a human body depends mainly on the following three factors :

1. Dry bulb temperature, 2. Relative humidity, and 3. Air velocity.

In order to evaluate the combined effect of these factors, the term *effective temperature* is employed. It is defined as that index which corelates the combined effects of air temperature, relative humidity and air velocity on the human body. The numerical value of effective temperature is made equal to the temperature of still (*i.e.* 5 to 8 m/min air velocity) saturated air, which produces the same sensation of warmth or coolness as produced under the given conditions.

## Comfort chart

The comfort chart shows the range for both summer and winter condition within which a condition of comfort exists for most people. For summer conditions, the chart indicates that a maximum of 98 percent people felt comfortable for an effective temperature of 21.6°C . For winter conditions, chart indicates that an effective temperature of 20°C was desired by 97.7 percent people. It has been found that for comfort, women require 0.5°C higher effective temperature than men. All men and women above 40 years of age prefer 0.5°C higher effective temperature than the persons below 40 years of age.

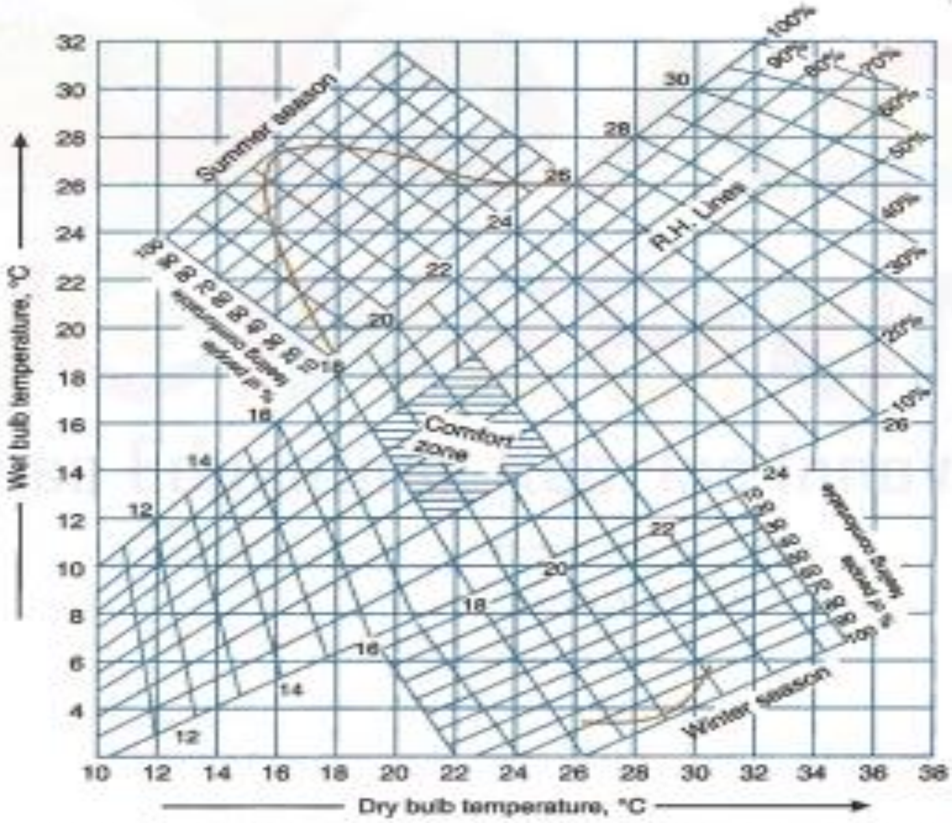



Fig. 17.1. Comfort chart for still air (air velocities from 5 to 8 m/min)

It may be noted that the comfort chart, as shown in Fig. 17.1, does not take into account the variations in comfort conditions when there are wide variations in the mean radiant temperature (MRT). In the range of 26.5°C, a rise of 0.5°C in mean radiant temperature above the room dry bulb temperature raises the effective temperature by 0.5°C. The effect of mean radiant temperature on comfort is less pronounced at high temperatures than at low temperatures.

The comfort conditions for persons at work vary with the rate of work and the amount of clothing worn. In general, the greater the degree of activity, the lower the effective temperature necessary for comfort.

Fig. 17.2 shows the variation in effective temperature with different air velocities. We see that for the atmospheric conditions of 24°C dry bulb temperature and 16°C wet bulb temperature correspond to about 21°C with nominally still air (velocity 6 m/min) and it is about 17°C at an air velocity of 210 m/min. The same effective temperature is observed at higher dry bulb and wet bulb temperatures with higher velocities. The case is reversed after 37.8°C as in that case higher velocities will increase sensible heat flow from air to body and will decrease comfort. The same effective temperature means same feeling of warmth, but it does not mean same comfort.

# Long & short questions chapter 6

1.

The pressure and temperature of a mixture of dry air and water vapour are 736 mm of Hg and 21°C. The dew point temperature of the mixture is 15°C. Determine the following using steam tables :

1. Partial pressure of water vapour ;
2. Relative humidity ;
3. Specific humidity ;
4. Enthalpy of mixture per kg of dry air ;
5. Specific volume of the mixture per kg of dry air.

**2**

A sample of air is having dry bulb temperature 21°C and relative humidity 30% at barometric pressure of 760 mm of Hg. Find : 1. Partial pressure of vapour ; 2. Specific humidity ; 3. Wet bulb temperature and corresponding saturation pressure ; 4. Precentage humidity or degree of saturation ; 5. Specific volume of dry air ; 6. Dew point temperature ; and 7. Enthalpy of moist air per kg of dry air.

Given : $R = 0.287$ kJ/kg K ; $c_p$(dry air) = 1.005 kJ/kg K ; specific heat of superheated vapour = 1.884 kJ/kg K and latent heat of vaporisation at dew point temperature = 2493 kJ/kg. Do not use psychrometric chart. Psychrometric tables can be used.

3.Write short notes on SLING PSYCHOMETER.
4.Explain cooling & dehumidification process.
5.Explain Comfort Chart .
6.a.What is effective temperature?
b.What is humidification & dehumidification?
c.What is relative humidity?
d.What is Gibbs_Daltons law?
e.What is the formula for Vapour density of moist air?
f.What is sensible cooling of air?
g.What is sensible heating of air?
h.What are the factors affect the comfort conditions?

# Chapter-7 Air conditioning system

## Factors Affecting Human Comfort

In designing winter or summer air conditioning system, the designer should be well conversant with a number of factors which physiologically affect human comfort. The important factors are as follows :

1. Effective temperature, 2. Heat production and regulation in human body, 3. Heat and moisture losses from the human body, 4. Moisture content of air, 5. Quality and quantity of air. 6. Air motion, 7. Hot and cold surfaces, and 8. Air stratification.

Factors affecting comfort air conditioning

1. *Temperature of air.* In air conditioning, the control of temperature means the maintenance of any desired temperature within an enclosed space even though the temperature of the outside air is above or below the desired room temperature. This is accomplished either by the addition or removal of heat from the enclosed space as and when demanded. It may be noted that a human being feels comfortable when the air is at 21°C with 56% relative humidity.

2. *Humidity of air.* The control of humidity of air means the decreasing or increasing of moisture contents of air during summer or winter respectively in order to produce comfortable and healthy conditions. The control of humidity is not only necessary for human comfort but it also increases the efficiency of the workers. In general, for summer air conditioning, the relative humidity should not be less than 60% whereas for winter air conditioning it should not be more than 40%.

3. *Purity of air.* It is an important factor for the comfort of a human body. It has been noticed that people do not feel comfortable when breathing contaminated air, even if it is within acceptable temperature and humidity ranges. It is thus obvious that proper filtration, cleaning and purification of air is essential to keep it free from dust and other impurities.

4. *Motion of air.* The motion or circulation of air is another important factor which should be controlled, in order to keep constant temperature throughout the conditioned space. It is, therefore, necessary that there should be equi-distribution of air throughout the space to be air conditioned.

## Equipments Used in an Air Conditioning System

Following are the main equipments or parts used in an air conditioning system :

1. *Circulation fan.* The main function of this fan is to move air to and from the room.

2. *Air conditioning unit.* It is a unit which consists of cooling and dehumidifying processes for summer air conditioning or heating and humidification processes for winter air conditioning.

3. *Supply duct.* It directs the conditioned air from the circulating fan to the space to be air conditioned at proper point.

4. *Supply outlets.* These are grills which distribute the conditioned air evenly in the room.

5. *Return outlets.* These are the openings in a room surface which allow the room air to enter the return duct.

6. *Filters.* The main function of the filters is to remove dust, dirt and other harmful bacteria from the air.

## Classification of Air Conditioning Systems

The air conditioning systems may be broadly classified as follows :

**1. *According to the purpose***

(*a*) Comfort air conditioning system, and

(*b*) Industrial air conditioning system.

**2. *According to season of the year***

(*a*) Winter air conditioning system,

(*b*) Summer air conditioning system, and

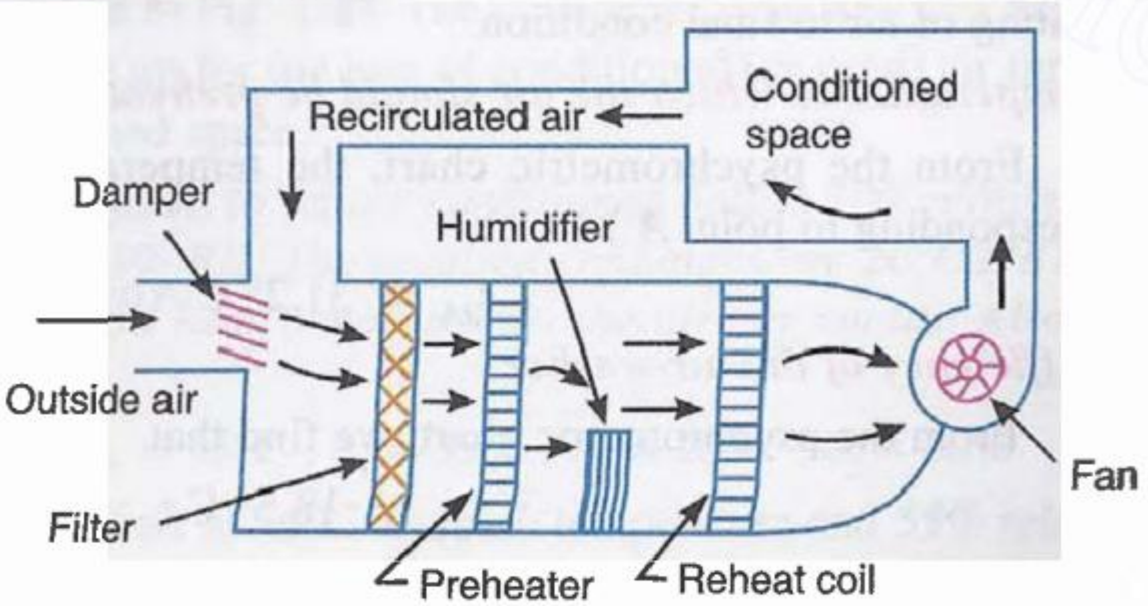
(*c*) Year-round air conditioning system.

**3. *According to the arrangement of equipment***

(*a*) Unitary air conditioning system, and

(*b*) Central air conditioning system.

## Winter Air Conditioning System



In winter air conditioning, the air is heated, which is generally accompanied by humidification. The schematic arrangement of the system is shown in Fig. 18.3.

The outside air flows through a damper and mixes up with the recirculated air (which is obtained from the conditioned space). The mixed air passes through a filter to remove dirt, dust and other impurities. The air now passes through a preheat coil in order to prevent the possible freezing of water and to control the evaporation of water in the humidifier. After that, the air is made to pass through a reheat coil to bring the air to the

designed dry bulb temperature. Now, the conditioned air is supplied to the conditioned space by a fan. From the conditioned space, a part of the used air is exhausted to the atmosphere by the exhaust fans or ventilators. The remaining part of the used air (known as recirculated air) is again conditioned as shown in Fig. 18.3.

The outside air is sucked and made to mix with recirculated air, in order to make up for the loss of conditioned (or used) air through exhaust fans or ventilation from the conditioned space.

**Example 18.3.** *Air at 10°C DBT and 90% RH is to be brought to 35°C DBT and 22.5°C WBT with the help of winter air conditioner. If the humidified air comes out of the humidifier at 90% RH, draw the various processes involved on a skeleton psychrometric chart and find : 1. the temperature to which the air should be preheated, and 2. the efficiency of the air-washer.*

**Solution.** Given : $t_{d1} = 10°\text{ C}$ ; $\phi_1 = 90\%$ ; $t_{d2} = 35°\text{C}$ ; $t_{w2} = 22.5°\text{C}$

First of all, mark the initial condition of air at 10°C dry bulb temperature and 90% relative humidity on the psychrometric chart as point 1, as shown in Fig. 18.4. Now mark the final condition of air at 35°C dry bulb temperature and 22.5°C wet bulb temperature, as point 2. Since the final condition of air is obtained with the help of a winter air conditioner, therefore the processes involved are as follows :

1. Preheating of entering air in a preheater,
2. Humidification of preheated air in a humidifier or air-washer, and
3. Reheating of humidified air in a reheater.

These processes are shown by 1-*A*, *A*-*B*, and *B*-2 respectively on the psychrometric chart as shown in Fig. 18.4. In order to obtain these processes on the psychrometric chart, draw a horizontal line through point 1 to represent sensible heating of air and from point 2 draw a horizontal line to intersect 90% relative humidity line at *B*. Now from point *B*, draw a constant wet bulb temperature line which intersects the horizontal line drawn through point 1 at point *A*. Now line 1-*A* represents preheating of air, line *A*-*B* represents humidification and line *B*-2 represents reheating of air to final condition.

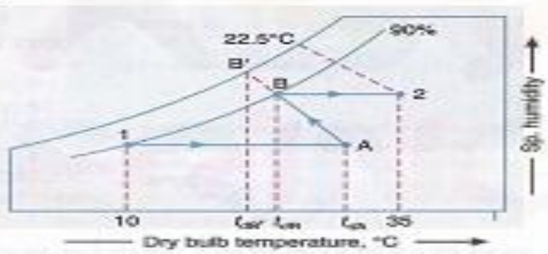


Fig. 18.4

**1.** ***Temperature to which the air should be preheated***

From the psychrometric chart, the temperature to which the air should be preheated (corresponding to point *A* ) is

$$t_{dA} = 31.2°\text{ C} \quad \textbf{Ans.}$$

**2.** ***Efficiency of the air-washer***

From the psychrometric chart, we find that

$$t_{dB} = 18.5°\text{ C}; \quad \text{and} \quad t_{dB'} = 17.5°\text{ C}$$

We know that efficiency of the air-washer

$$= \frac{\text{Actual drop in DBT}}{\text{Ideal drop in DBT}} = \frac{t_{dA} - t_{dB}}{t_{dA} - t_{dB'}}$$

$$= \frac{31.2 - 18.5}{31.2 - 17.5} = 0.927 \text{ or } 92.7\% \quad \textbf{Ans.}$$

# Summer Air Conditioning System

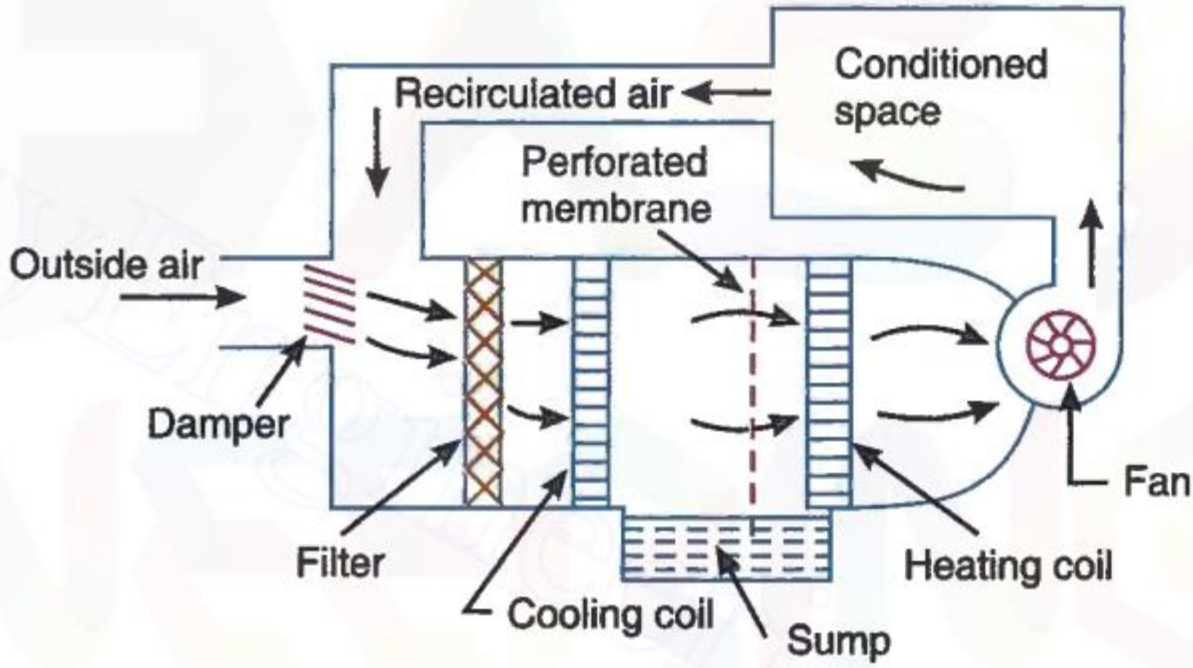



**Fig. 18.5.** Summer air conditioning system.

The outside air flows through the damper, and mixes up with recirculated air (which is obtained from the conditioned space). The mixed air passes through a filter to remove dirt, dust and other impurities. The air now passes through a cooling coil. The coil has a temperature much below the required dry bulb temperature of the air in the conditioned space. The cooled air passes through a perforated membrane and loses its moisture in the condensed form which is collected in a sump. After that, the air is made to pass through a heating coil which heats up the air slightly. This is done to bring the air to the designed dry bulb temperature and relative humidity.

Now the conditioned air is supplied to the conditioned space by a fan. From the conditioned space, a part of the used air is exhausted to the atmosphere by the exhaust fans or ventilators. The remaining part of the used air (known as recirculated air) is again conditioned as shown in Fig. 18.5. The outside air is sucked and made to mix with the recirculated air in order to make up for the loss of conditioned (or used) air through exhaust fans or ventilation from the conditioned space.

**Example 18.4.** *The amount of air supplied to an air conditioned hall is 300 m³/min. The atmospheric conditions are 35°C DBT and 55% RH. The required conditions are 20°C DBT and 60% RH. Find out the sensible heat and latent heat removed from the air per minute. Also find sensible heat factor for the system.*

**Solution.** Given : $v_1 = 300$ m³/min ; $t_{d1} = 35$°C ; $\phi_1 = 55\%$ ; $t_{d2} = 20$°C ; $\phi_2 = 60\%$

First of all, mark the initial condition of air at 35°C dry bulb temperature and 55% relative humidity on the psychrometric chart at point 1, as shown in Fig. 18.6. Now mark the final condition of air at 20°C dry bulb temperature and 60% relative humidity on the chart as point 2. Locate point 3 on the chart by drawing horizontal line through point 2 and vertical line through point 1. From the psychrometric chart, we find that specific volume of air at point 1,

$$v_{s1} = 0.9 \text{ m}^3\text{/kg of dry air}$$

∴ Mass of air supplied,

$$m_a = \frac{v_1}{v_{s1}} = \frac{300}{0.9} = 333.3 \text{ kg / min}$$

*Sensible heat removed from the air*

From the psychrometric chart, we find that enthalpy of air at point 1,

$$h_1 = 85.8 \text{ kJ/kg of dry air}$$

Enthalpy of air at point 2,

$$h_2 = 42.2 \text{ kJ/kg of dry air}$$

and enthalpy of air at point 3,

$$h_3 = 57.4 \text{ kJ/kg of dry air}$$

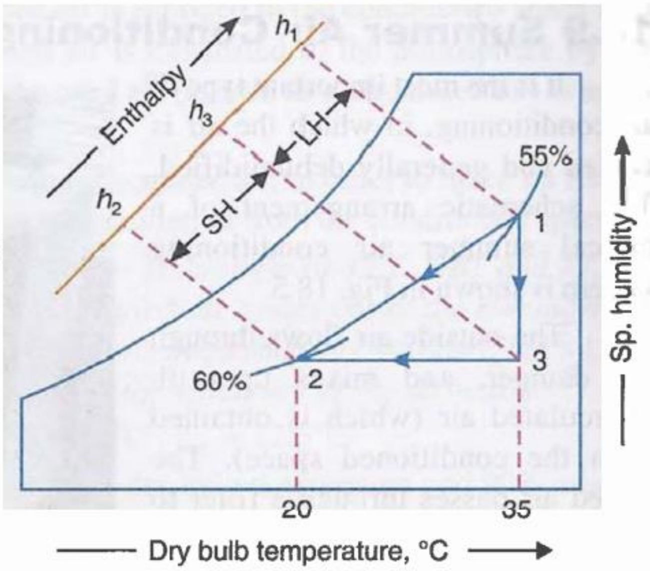



Fig. 18.6

We know that sensible heat removed from the air,

$$SH = m_a (h_3 - h_2)$$
$$= 333.3\,(57.4 - 42.2) = 5066.2 \text{ kJ/min } \textbf{Ans.}$$

*Latent heat removed from the air*

We know that latent heat removed from the air,

$$LH = m_a (h_1 - h_3)$$
$$= 333.3\,(85.8 - 57.4) = 9465.7 \text{ kJ/min } \textbf{Ans.}$$

*Sensible heat factor for the system*

We know that sensible heat factor for the system,

$$SHF = \frac{SH}{SH + LH} = \frac{5066.2}{5066.2 + 9465.7} = 0.348 \textbf{ Ans.}$$

## Long and short questions chapter 7